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THESIS

**ASSESSMENT OF DIESEL ENGINE CONDITION
USING TIME RESOLVED MEASUREMENTS AND
SIGNAL PROCESSING**

by

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September 1996

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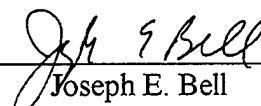
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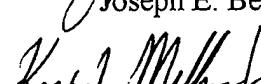
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ABSTRACT

An experimental investigation was conducted to access methods of detecting, and localizing faults in a diesel engine. A three cylinder, two stroke Detroit 3-53 engine was heavily instrumented for time resolved measurements. In particular, a 3,600 count per revolution optical encoder was used along with accelerometers mounted on various engine structures, in-cylinder pressure measurements and a variety of steady state sensors, such as exhaust temperatures. A large number of baseline data were taken to establish the statistical characteristics on the signals from the engine. These runs were followed by a series of experiments where the cylinder head assembly bolt torque were varied parametrically. Standard spectral analysis and Joint Time Frequency Analysis (JTFA) were used to identify the fundamental vibration characteristics of the engine. The vibration frequencies were checked for consistency against first order models of the engine assembly and reasonable agreement was found. In addition, a new technique for accessing engine health using time of arrival of encoder signals was investigated.

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I. INTRODUCTION

Diesel engines are widely utilized for both military and civilian applications. They are used for the propulsion of ships, boats, trucks, automobiles, and locomotives, and as prime mover for generators, pumps, etc. These applications require power ranging from less than one horsepower to tens of thousands of horsepower. The U.S. Navy uses diesels for propulsion and power on surface ships and as auxiliary generators on submarines.

The Navy's need for readiness requires that these diesel engines should have a high availability. Therefore, these engines must meet standards and are subject to many inspections and operational tests. However, the price of this readiness is costly in terms of money, time, redundancy, and down time. Furthermore, despite the long existence of diesel engines, the amount of maintenance required for a diesel is nearly three times that of a gas turbine in terms of maintenance dollar per horsepower hour. [Ref. 1]

The U.S. Navy currently uses a system of maintenance called Diesel Engine Trend Analysis (DETA) for their diesels. This is a trend based, steady state, exhaust temperature, power, etc system. DETA is used with the Preventative Maintenance System (PMS) which is based on Regularly Scheduled Maintenance (RSM). The engines are periodically broken down for inspections and servicing based on engine operating hours, or time since the last maintenance. Regularly Scheduled Maintenance is based on a statistical measure of when parts fail versus time and the choice of time interval reflects a probable time in which a very small fraction of parts will fail in service. RSM is based on a predicted time to failure of "the weakest link" in the system. Thus, the servicing of most engines results in the replacement

of predominately good engine parts and the temporary loss of that engine for the duration of the maintenance. This is costly in terms of labor, parts and equipment down time.

Another maintenance philosophy is Conditioned Based Maintenance (CBM). This method uses the measurement of certain parameters to gauge the condition of an engine. Measurements of specific parameters of an engine are taken and in conjunction with a knowledge/data base of that engine or class of engines a determination is made concerning the requirement for servicing. Ideally, CBM would predict and localize faults in a system before they become serious and prevents replacement and tear down of a well running engine.

Comparing CBM to RSM, CBM requires far fewer maintenance actions and the number of service failures are also reduced. CBM if properly implemented can lead to significant savings as well as increased system readiness.

Implementing a CBM system on reciprocating engines especially internal combustion is more difficult than for similar classes of rotating turbomachines. Internal combustion engine diagnosis is a complex task, for numerous reasons: very unsteady process, more complex engine geometry, noisy data, unavailability of a complete and faithful model, multiple faults. Finally, little work has been done in this area compared to rotating machines.

This thesis investigates predicting engine health utilizing two different techniques. One utilizing vibration transducers (accelerometers) on various locations of the engine to predict changes in engine health based on analysis of these signals with spectral analysis and Joint Time Frequency Analysis (JTFA). Vibration signals were chosen for analysis because the sensors are cheap, easy to install, and should be a good indicator of the condition of the engine. Vibration analysis does not drastically change the characteristics of the engine and

would be relatively easy to implement even in an existing engine. These are all key points if CBM is to be implemented. Vibration signals of the engine were analyzed while varying the cylinder head bolt torques.

Another technique was investigated which used the shaft speed of the diesel as a predictor. Through the course of a revolution of the shaft, the engine undergoes the firing of all cylinders of the engine. These torque pulsations create simultaneous changes in the rotational speed of the shaft. These changes may be a good predictor of certain classes of engine faults.

Another system used was an Engine Cycle Analyzer for a thermodynamic investigation of the engine.

The objectives of this thesis are:

1. To instrument a diesel engine with various high speed sensors. Establish a baseline of data and utilize the data collected in conjunction with spectral analysis software to analyze the signals for predictions of engine health.
2. Establish the statistical nature of the signals repeatability, and the number of cycles needed for a meaningful average.
3. Determine the impact that a variation in cylinder head bolt torque has on the vibration signatures of the engine.
4. Establish safe operating instructions and control of the engine and its auxiliary equipment for further research and thermodynamic laboratories.

Chapter II presents the current techniques used in the monitoring and diagnosis of diesel engines for conditioned based maintenance. It reflects some of the "state of the art"

in condition based diagnosis of diesel engines.

Chapter III provides a brief description of the diesel engine and it's dynamometer and test stand. The instrumentation installed on the engine and other modifications are also discussed. It also discusses the Superflow Engine Cycle Analyzer (ECA) software used.

Chapter IV discusses the results of the various experiments performed on the engine. Section 4.1 discusses the Engine Cycle Analyzer (ECA) results. These data are primarily a thermodynamic investigation of one cylinder throughout the engine cycle. Section 4.2 discusses the effects of varying head bolt torques and the response in the vibrational analysis of the to help predict engine faults. Further, it discusses the repeatability of engine signals and the validity of these techniques for an engine health predictor. Section 4.3 reveals the results of analyzing the time differential between signals of an optical encoder and its possibilities in the field of engine diagnostics.

Chapter V contains conclusions from this study and recommendations for further research and possible refinement of the techniques tested.

II. STATE OF THE ART AND REVIEW OF LITERATURE

2.1 CURRENT EXPERT AND AUTOMATED SYSTEMS

There are several automated or expert systems available to diesel users. These systems use a variety of sensors and algorithms to predict engine health. Most of these sensors are steady measurements, and a complete physical understanding of the engine process is missing. There does not appear to be a single sensor that provides a complete picture of the condition of a diesel engine, however using many sensors apparently many faults can be isolated.

One such expert system is called the Diesel Trap and is manufactured by the Beta Monitors & Controls. This system may be used with virtually all high-, medium-, and low-speed diesel engines. It may be used as a monitor for both performance analysis and for mechanical condition monitoring. [Ref. 2]

The Diesel-Trap is a PC based software package which also consists of a data collector and a variety of sensors which can monitor overall and individual cylinder power output; combustion and injection timing; maximum cylinder pressure and cylinder pressure as a function of crankshaft position; engine balance or deviations between cylinders and developing performance trends. It allows historical comparisons and current trends. And offers assessments of valves, injectors, pumps, rings, cylinder liners, bearings and turbochargers. [Ref. 2]

Another expert system is currently being developed for the Canadian Coast Guard. It is a system composed of three predictive techniques; oil analysis, vibration analysis, and the

performance analysis software. Performance analysis software may be used for both current data and a comparison to historical data. [Ref. 3]

The oil analysis and vibration analysis are very conventional programs, but the performance analysis software systems (PASS) is an automated monitoring and analysis system to contribute to the predictive condition based maintenance of ship's machinery. It is user friendly and robust software which provides trend analysis and graphical analysis of historical condition/performance data versus time and their deviations from baselines; and includes a predictive extrapolation capability to estimate time to alarm. [Ref. 3]

Another general purpose engine analyzer is being developed by the University of Michigan. This systems can analyze idle speed stability and cylinder power balance, misfire detection, compression checks, fuel diagnosis, O2 sensor and catalyst diagnostics. It relies on the rotation of the crankshaft for an overall estimate performance based on the measurement of instantaneous crankshaft angular velocity. The system also uses a Failure Detection and Isolation Theory (FDI) to generate an estimate of the engines state by anticipating the output of a given input and generates a discrepancy if its expected output is not met. [Ref. 4]

These are just examples of three expert systems. There are others that are commercially available.

2.2 LITERATURE REVIEW

Research is being conducted in the field of engine diagnostics based on deviations of shaft speed. It is based on the acceleration and deceleration of the shaft during normal operations due to the compression and expansion in the pistons. Enikeev et al, described this

technique for the diagnosis of a diesel generator. [Ref. 5]

Mauer uses this method of deviations of shaft speed and details development of his model for the diagnosis of an engine. His work uses front-end and flywheel engine speeds for deriving the individual cylinder performance coefficients. He shows the effects of low and high speed and discusses the isolation of individual cylinders for diagnosing multi-cylinder engines. [Ref. 6]

Shiao and Moskwa utilized nonlinear Sliding Observers in conjunction with shaft speed variations to estimate cylinder pressure and heat release rates in a running engine. This replaces the need for expensive and frail pressure measuring devices to be installed in the engine. This data can then be used in a variety of method for engine diagnostics. [Ref. 7]

Some methods on isolation and identifying faults with in an engine are discussed by Laukonen et al. [Ref. 8] This paper constructs a method for computer diagnosis based on multiple sensors. It uses a Failure Detection and Isolation (FDI) system to characterize faults. It is much like the system used by a University of Michigan group [Ref. 4] discussed earlier.

An engine cycle analyzer and simulator for a personal computer was created by Chen et al. [Ref. 9] This group uses multiple sensors, but focus on in cylinder pressure sensors for their analyzer. They discuss the thermodynamic aspects, cyclic performance and diagnosis in this extensive work.

2.3 JOINT TIME FREQUENCY ANALYSIS OF SIGNALS

Many problems in machinery diagnostics are characterized by frequency content that varies considerably and regularly with time [Ref. 10]. This is a problem since signals have,

traditionally, been analyzed in either the time or the frequency domain, but not jointly in time and frequency. For signals that do not change in their spectral content in time, standard spectral analysis reveals what frequencies and phase are present and their relative intensities.

[Ref. 11]

Many signals, however, change frequency content over time. Fourier Analysis does not show when those frequencies occurred in time. The aim of joint time-frequency analysis (JTFA) is to describe and determine how the frequencies in non-stationary signals change over time. [Ref. 11]

Below, Figure 2.1, is an example of a JTFA plot. The lower middle block is the raw signal. It may be any type of saved data. The block on the right under the spectrum sign is the Fourier Transform (FFT) of the signal. This shows the base frequencies which make up the signal, but as is the weakness of traditional analysis does not tell when a certain frequency started or stopped. The largest block is the JTFA of the signal. The four “island” represent the frequency and the times when they started and stopped. The ordinate is frequency and the island may be compared to the FFT to the right. The abscissa is the time that the specific frequency started and stopped. The usefulness of this technique is evident by comparing the FFT to the JTFA, as the second occurrence in the signal can not be distinguished from the first occurrence in the FFT.

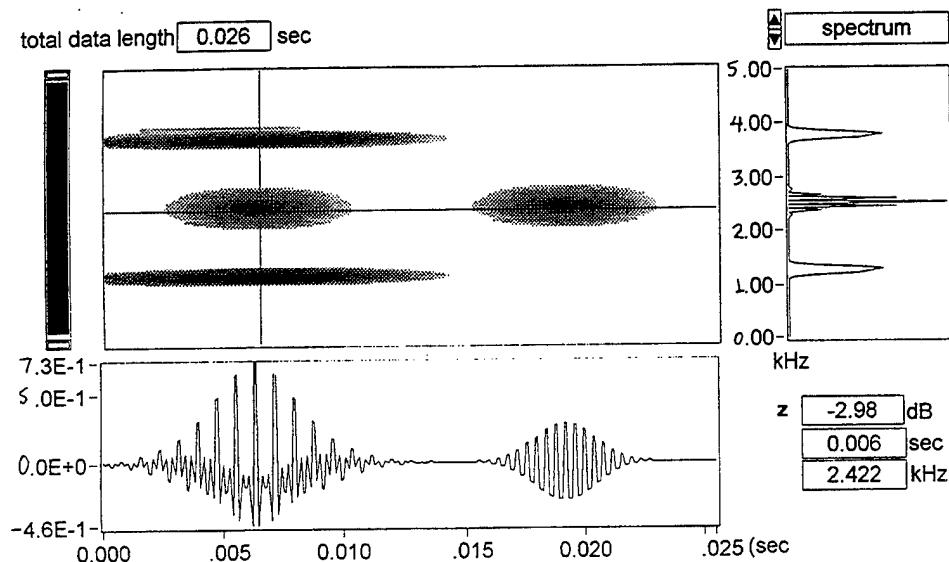


Figure 2.1. Joint Time Frequency Plot of a Gaussian Signal.

Thus, for more complicated signals with several frequency components and varying on and off times, the JTFA is particularly useful.

There are several methods of displaying the information in the JTFA window. They are Wigner-Ville, Cone Shaped, Choi-Williams, Short-Time Fourier Transform, Gabor Spectrogram, and Adaptive Spectrogram. These methods are different algorithms and each has its own strengths and weaknesses.

The Wigner-Ville, Cone-Shaped, and Choi-Williams distributions are all related and are special cases of bilinear transforms. The Wigner-Ville distribution (WVD) is a Fourier transform of the time-dependent autocorrelation function $Z_s(I+m)Z_s(I-m)$. It provides both time and frequency information in the same plot, however cross terms cause considerable interference [Ref. 11]. This interference can make it difficult to determine exactly when a frequency component starts. Rizzoni used WVD and Choi-Williams for the detection of

knock in internal combustion engines. [Ref. 12] WVD has been used by Choi to predict faults in gear transmission systems under normal operating conditions successfully [Ref. 13]. It has also used by Gaberson to locate the angular position of the impact or discontinuity associated with gear tooth faults, but the interference causes problems with interpretation [Ref. 10].

The Cone-Shaped distribution is very similar to the Wigner-Ville, yet it uses a smoothing function. This smoothing eliminates some of the interference experienced by Wigner-Ville. The cone shaped distribution can also be negative which on physical systems is not desirable. [Ref. 11]

Choi-Williams distribution (CWD), like the Cone-shaped distribution, also has a smoothing function. The smoothing usually eliminates interference, however requires extra computing time. The distribution may also be negative[Ref. 11]. CWD is recommended for its detail and significant structure in the frequency plane by Gaberson.[Ref. 10]

Short-Time Fourier Transform is a common method of JTFA. It uses fast Fourier transforms (FFT) computed over small slices of the signal. The FFT computes the frequency spectrum for each slice of the data and only the square magnitude of the FFT is kept for the STFT. This method is positive, but its resolution in both time and frequency is inferior to the WVD or CWD.[Ref. 11 and Ref. 10] STFT is used in speech analysis, but does not provide sufficient resolution for machinery diagnostics [Ref. 10]. However, it is a popular machinery diagnostics analysis for start up and shutdown and is referred to by signal analyzer manufacturers as a waterfall function [Ref. 10].

Gabor Spectrogram represents the time-dependent power spectrum of a signal in terms of a series of time-frequency functions. The Gabor spectrogram can converge to a

WVD. Higher order Gabor spectrograms have good resolution, but more cross term interference; lower order Gabor spectrograms have less interference but also lower resolution. It requires more computing time, but offers better resolution than STFT, and less interference than WVD, CWD, and Cone -shaped. [Ref. 11]

Adaptive spectrogram adjusts the variance, time, and frequency centers of the Gaussian functions to best match the analysis signal. It has the best resolution of all the techniques and has little cross term interference. This method uses more computing time and is best suited for quasi stationary signals. [Ref. 11]

Adaptive was chosen for this research for its clarity. Some of the algorithms compute displays which are very difficult to categorize. They appear to have “tiger stripes” or some other strange patterns. The Adaptive distribution displays almost individual islands for the different frequencies and this display seems the simplest to understand.

Joint Time Frequency Analysis is of particular use for engine and machinery analysis. Reciprocating engine signals change through a cycle. Unlike turbo machines, reciprocating engines are involved with the non uniform process of compression, combustion, and expansion. This unsteady event causes many different signals to be generated, and all at different times over a cycle. These frequency shifts, or the times they occur, may be indicative of a condition or a problem in the engine.

III. EXPERIMENTAL SETUP

3.1 THE ENGINE

3.1.1 Overview

The diesel engine used for this research is a Detroit Diesel Series 53 engine model 5033-5001N. It is a three cylinder naturally aspirated two stroke engine which was formerly installed in a U. S. Army 1-1/4 ton 6X6 Cargo Truck. This vehicle was phased out by the Army several years ago, but this engine type is still manufactured and used in industrial and marine applications. The engine is a representative example of a typical diesel engine.

Table 3-1 Engine Characteristics [Ref.14]

Model	5033-5001N
Engine Type	In line-2 Cycle-Naturally Aspirated
Number of Cylinders	3
Bore and Stroke	3.875 x 4.50 inches
Exhaust Valves per Cylinder	4
Engine Displacement	159 cubic inches (2.61 liters)
Compression Ratio	21:1
Maximum Power Output	92 bhp
Full Load Speed	2800 RPM
Peak Torque	198 lb-ft
Peak Torque Speed	1500 RPM
BMEP	83 lb/in ²

3.1.2 Cylinder Head Assembly

Access to the cylinders is limited without extensive modification to the engine's cylinder head assembly or cylinder liners. However, in order to gain cylinder pressure information pressure transducers must have access to the cylinder. This problem was solved by installing a cylinder head assembly from a similar but different engine which allows cylinder access through glow plug mounting holes. This modification allowed the mounting of pressure transducers in the cylinders.

The cylinder head assembly is secured to the engine cylinder block by eight 9/16 inch diameter bolts. These bolts are tightened to 170-180 ft-lbs torque.

3.1.3 Engine Orientation Conventions

Some orientation conventions for the engine were chosen for purposes of consistent labeling. The front of the engine has the crankshaft pulley, the balancing wheel and the fan belts and wheels. The back of the engine has the flywheel and the power take-off shaft. The cylinders are numbered from front(1) to back(3). The firing order of the cylinders is 1-3-2 for this right hand engine. The axes for the engine is considered as the following: X axis runs from the front to the back of the engine, Y axis runs vertically through the engine (with gravity), and the Z axis runs from left to right when facing the front of the engine, as in Figure 3.1.

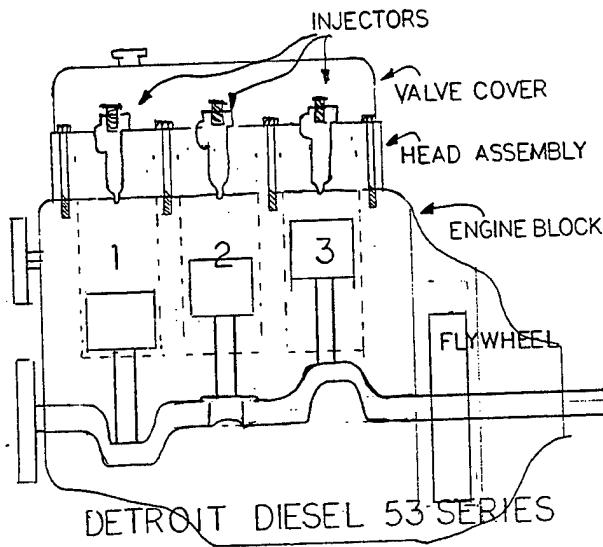


Figure 3.1 Detroit Diesel 3-53 with labeling convention. The cylinders are from left to right, 1-2-3.

3.1.4 Top Dead Center (TDC)

The data collection system needs a reference which will remain constant. Time is not a good reference as the engine changes speed with varying torque. This drift in speed would create sampling difficulties. The reference chosen was Top Dead Center (TDC) of cylinder number one. TDC occurs when the volume of the cylinder is a minimum; it is the highest point of travel of the piston. TDC is used by the optical encoder, which will be explained later, to reference its angular measurements. The procedure used to establish TDC is outlined in Appendix G.

3.2 DYNAMOMETER TEST STAND

The engine is controlled by a SuperFlow 901 Engine Dynamometer. This acts as both the control console and as the engine's power absorber. The absorber is a computer

controlled water brake. This system is adequate for diesel research, however an electric motoring dynamometer may work better.

Through out most of this research, the computer system controlling the automatic throttle function of the diesel was inoperable. This continues to be a problem, however it does not affect steady state operations. It is believed to be the servo control card loosening up.

Instructions and operations of the engine and test stand and dynamometer are included in detailed instructions in Appendix A. These instructions apply to basic engine operation and all ancillary equipment.

3.3 ENGINE CYCLE ANALYZER

A SuperFlow Engine Cycle Analyzer (ECA) is also installed on the engine. This system is a PC Based computer which monitors various functions of the engine and allows data to be gathered. The ECA has two basic parts. The first part is an analog to digital converter; the second part is the ECA software.

The analog to digital converter digitizes a voltage signal from an accelerometer or pressure transducer. This digitized signal may be retained as the actual voltage or converted from a voltage to the sensors measuring units. The 12 bit converter requires that the sensors have a plus or minus five volt ($\pm 5V$) signal. This provides a digital resolution of 2.44mV.

The analog to digital converter uses an optical encoder as a trigger and a clock for data collection. It may be programmed to take data each time a pulse from the optical encoder is received, or some multiple of that number.

The ECA allows several setup programs to be used for data acquisition and allows up to four channels of data to be taken at one time. One of the channels is usually pressure data, the remaining three channels for this experiment were vibration data. The setup program allows the user to set gain factors, units and sensor voltage offsets. These are described in Appendix C.

The ECA will collect from 1 to 999 cycles of data for the user. Diesel have a cycle to cycle variation as well as cylinder to cylinder variation. This creates problems if too few cycles are examined. Noise and unsteady conditions are hard to eliminate and may be considered significant events. A problem with collecting too many cycles of data is that the engine may change conditions slightly during this acquisition time.

The other part of the ECA is the engine thermodynamic data processor. It uses pressure data from a cylinder to compute various thermodynamic conditions in the engine cycle. This feature is discussed in more detail in Section 4.1.

3.4 INSTRUMENTATION

3.4.1 Optical Encoder

Top Dead Center (TDC) is the reference chosen against which all events are measured. It and the engine's angular speed are monitored by an optical encoder. The optical encoder provides a TTL (transistor, transistor logic) output of zero to five volts. All measurements were conducted with a 3600 count per revolution encoder which provides resolution of 0.1° per pulse. Another encoder is available which has 720 counts per revolution which has 0.5° per pulse. Another five volt signal is sent by the encoder every

time it passes TDC.

The crankshaft pulley was chosen for mounting the encoder for the following reasons: it is easily accessible, turns directly with the crankshaft, has a face which may be altered for a mounting sleeve, and it has many mounting bolts in the vicinity on which to bolt the mounting bracket to the engine.

An adaptor plate of aluminum was constructed to mount over the crankshaft pulley. This adaptor has a sleeve with two set screws to accept the shaft of the encoder. The plate is bolted to the pulley by two 3/8 inch bolts and has two dowels ninety degrees apart to ensure proper mounting.

The crankshaft pulley and the adaptor were removed from the engine and placed in a milling machine. This procedure was to ensure that the pulley, the adaptor plate, and it's center driving sleeve all turned concentrically. This was completed to a tolerance of 0.0005 inches.

The encoder needed a new mounting bracket. The former mounting bracket was a hanging type which did not provide adequate support for the encoder bearings. It also allowed the encoder to move too much during engine operations. The encoder requires very fine tolerances for run-out, misalignment and shaft-load [Ref. 15]. Table 3-2 describes the tolerances.

TABLE 3-2 Heidenhain Optical Encoder Alignment Requirements

Shaft Radial Run out	0.003 inches
Shaft Misalignment	0.005 inches
Angular Error	$\pm 1^\circ$
Shaft Axial Run out	0.005 inches

The new encoder mount was manufactured from equal lengths of brass rod and aluminum plate. The rods were drilled and tapped to accept a length of all-thread which matched existing bolt holes in the cylinder block. These rods were tightened to the engine and transfer/marker screws were utilized to mark the exact locations of the rods center on the aluminum plate. These marks were drilled to accept mounting screws.

With the crankshaft pulley adaptor in place and a transfer/marker screw installed in the adaptor sleeve, the center hole for the encoder was marked on the plate. The aluminum plate was drilled and milled to accept the encoder and to firmly mount the encoder to the plate.

All of the tolerances for manufacture were kept as fine as possible and final assembly could not be completed unless all part were in alignment.

A flexible coupling is available for the encoder and eliminates the need for extreme tolerances in the mounting procedure, however it was not used. The coupling would allow too much torsional spring effects and make the encoder useless during large torque or RPM changes. Figure 3.2 is a photograph of the current optical encoder mounting.



Figure 3.2 Heidenhain 3600 Count Optical Encoder and Mounting Bracket

The encoder mounted on the engine setup is a 3600 count Heidenhain encoder. The encoder which came with the Engine Cycle Analyzer (ECA) was the Heidenhain Model ROD 428D.0003-3600 which broke during engine operations. It was replaced by a Heidenhain Model ROD 426.000B-3600. This replacement model is adequate for the task. The only difference between the two encoders is some self test circuitry which is not available through the ECA.

The encoder provides a five volt TTL output. Its frequency is 3600 times that of the engine. Since this signal may be a diagnostic tool for the engine's condition care was exercised in its installation. This use of the encoder was investigated and is explained in Chapter 4.3.

3.4.2 Pressure Transducers

Pressure transducers are required by the Engine Cycle Analyzer (ECA) for thermodynamic data. There are some problems with pressure sensor mounting in a running

engine. The frequency response of the transducer must be great enough to resolve 0.1° of crank angle. The current generation of piezoelectric pressure transducers are good for this task, however they still have some difficulty due to the extreme operations conditions in the engine.

The preferred installation of a pressure transducer when possible is water cooled, flush to the piston bowl, reliable and linear, and heat shielded to minimize flame impingement effects [Ref 9]. An installation meeting all of these attributes could not be accomplished by this facility without extensive alterations to the engine.

The pressure transducers mounted in this facility are Kistler Internal Combustion Testing pressure transducers, Model 6125A11. This model uses a twin diaphragm and TiN coating to minimize thermal shock. The mounting threads match those of the engine's glow plugs. These transducers have the characteristics outlined in Table 3-3:

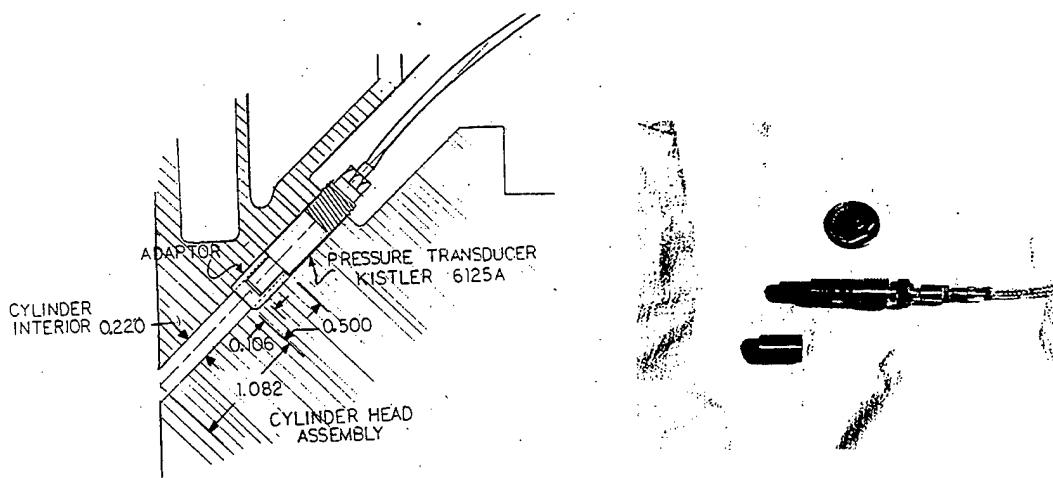
Table 3-3 Pressure Transducer Technical Data

Measuring range	0- 3850 psi
Threshold	0.03 psi
Natural Frequency	75 kHz
Linearity	$\pm 0.5\%$ FSO
Operating Temperature Range	-195 to 350 °C
Thermal Sensitivity	$\pm 1\%$ (200 ± 50 °C)

The current engine configuration does not easily permit the installation of flush mounted transducers, so adaptors had to be designed to accommodate their installation. The adaptor designed was very simple and provides adequate results.

The transducer casings are manufactured of 17-4PH stainless steel, so the adaptors were manufactured of the same material to ensure the thermal expansion would be the same. The adaptors are sleeves which are 0.500 inches long and are the same outer diameter as the pressure transducer body. The adaptor's inner diameter is 0.002 inches larger than the diaphragm section of the transducer. It extends beyond the diaphragm of the transducer by 0.1150 inches.

The adaptor is slipped over the transducer and mounted in the glow plug hole. Figure 3.3 shows the mounting setup and a transducer and adaptor.



a) Transducer mounting

b) Pressure transducer

Figure 3.3 Pressure Transducer and Mounting

This installation is still not optimal, due to the long void length from the cylinder to the diaphragm. This length is 1.193 inches long. This void can result in inaccuracies, but it does provide some protection from flame-induced thermal shock. These two problems are

the main sources of error for pressure transducers.

To alleviate the problem of thermal shock transducer mountings should have a small cavity to quench the flame before it reaches the transducer. The problem with this recess is that the passage will possess two resonant frequencies which will interfere with the measurement process. One resonance frequency is the Helmholtz resonance. The natural frequency of a Helmholtz oscillator is given by the following equation:

$$f_H = \frac{c}{2\pi} \sqrt{\frac{A_p}{V_c l_p}} \quad 3.1$$

Where c is the speed of sound in air, A_p is the cross-sectional area of the passage, V_c is the volume of the cavity, and l_p is the length of the passage. The Helmholtz resonance for the current transducer mounting is 7200 Hz. [Ref. 15]

The other resonance is caused by the excitation of a sonic standing wave in the passage. The natural frequency of a standing wave present on a passage with both ends open is given by Equation 3-2. [Ref. 15]

$$f_s = \frac{C}{l_p} \quad 3.2$$

This frequency is 14,850 Hz for the transducer mounting.

The current mounting system does have a long entrance passage. Kistler designed another adaptor for NPS which utilizes a Model 6053003 pressure transducer. Their adaptor uses a machined hollow copy of the installed glow plug with the transducer inserted inside.

This transducer/adapter combination would reduce the length of the passage to approximately 0.59 inches. This adaptor was not purchased, but it may increase some accuracy for pressure measurements.

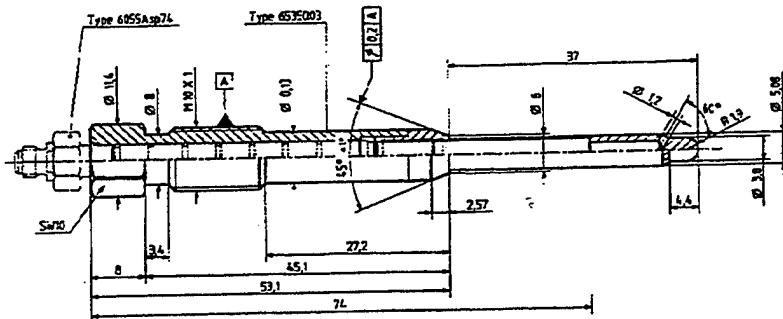


Figure 3.4 Pressure transducer/ Glow plug adaptor designed by Kistler utilizing a Model 6053B60

Piezoelectric pressure transducers work on the principle that a quartz element will exhibit a specific electrical charge as it is compressed. It is necessary to have a reference pressure to compare with the increased pressure to deduce the absolute pressure. Piezoelectric transducers are very good at indicating pressure difference but not an absolute pressure. The pressure in the engine cycle is 18.52 psi when the cylinder is admitting air. This 18.52 psi is the published air box pressure due to the air blower and is used by the pressure transducer as a reference against which to measure the pressure during the rest of the cycle. The pressure scale factor and the charge amplifier gain setting were computed by motoring the engine with a transducer installed and computing the pressure based on the known volume compressed with γ equal to 1.3. It was computed that the pressure in a nonfiring cylinder undergoing a 21 to 1 volumetric compression with air has a maximum

pressure of 725 psi. (Pressure curves from Detroit Diesel need to be obtained to verify this pressure.) This pressure was taken as the standard and used to calibrate the pressure transducer and charge amplifier pair.

3.4.3 Accelerometers

Nine accelerometers were installed on the engine cylinder head bolts. There are also three accelerometers mounted on the Cylinder Head assembly, the Cylinder Block, and the Engine Base. These accelerometers are small, low mass so as not to change the mass of the system, high frequency and high temperature, due to the conditions inside the Engine Valve Cover. They were mounted on Cylinder Head Bolts and on different major engine assemblies. The head bolts were chosen as the best location to mount acceleration transducers. The bolts provide a known location, symmetry and will be available on virtually all types of diesel engines. The eight bolts bracket the three cylinders, with four bolts around each cylinder; the middle cylinder sharing the inner bolts of the outer cylinders. The accelerometers mounted on the engine were mounted mainly for convenience. The sensors are orientated in the various planes in accordance with the following figure.

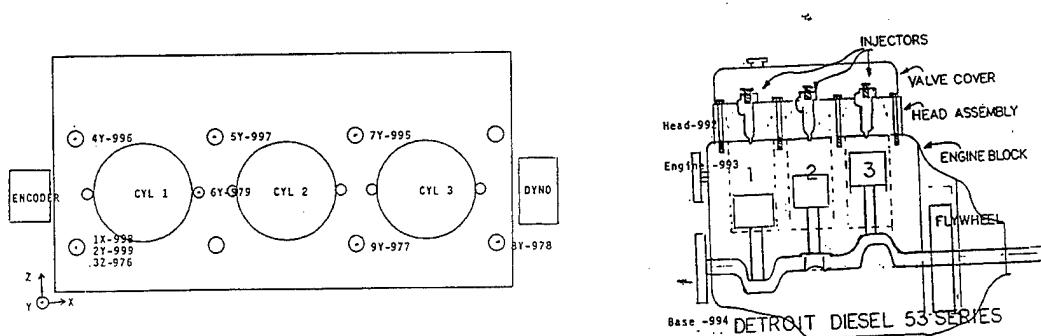


Figure 3.5 Accelerometer Locations (Sensor number, Axis of measure, Serial Number)

The figure shows the location of the bolts and the number of the sensor and its coordinate orientation. The three digit number following the sensor number and orientation is the serial number of the accelerometer. This is required for the calibration constant used by the software. A calibration certificate is provided with each accelerometer and the inverse of the voltage sensitivity per g of acceleration which is specific for each accelerometer. Appendix C contains a list of the appropriate calibration factors. The accelerometers calibration were checked with a one g calibration source to ensure the voltage sensitivity entered into the software is correct.

The accelerometers are mounted to tri-axial mounting blocks and the blocks are mounted to the engine head bolts or bolts which may be screwed into various bolt holes on the engine. The accelerometers are mainly oriented in the y direction (straight up), as it is believed to be the most useful direction, since this is the direction of the primary force.

IV. RESULTS

4.1 ENGINE CYCLE ANALYZER

The Engine Cycle Analyzer software uses data from the pressure transducers to investigate the thermodynamics of the engine. The data examined most was pressure versus theta, pressure versus volume, the log of pressure versus the log of volume, and temperature versus the entropy.

The pressure versus theta curve seen below, in Figure 4.1, is typical in shape for any sample taken on this running engine. The pressure magnitude increases with increasing RPM, but the shape remains constant. The maximum pressure for different runs at the same conditions tended to vary slightly but fundamentally remained the same.

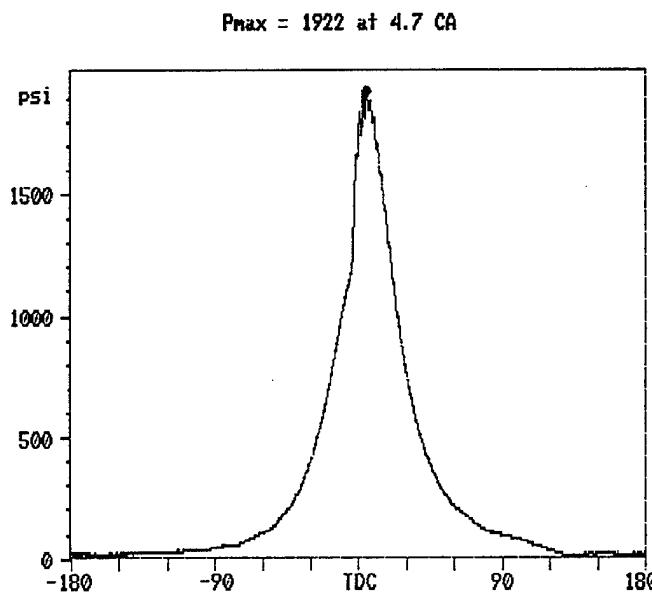


Figure 4.1 Pressure vs. Crank Angle Firing Engine (1800 RPM 125 ft-lb)

The following diagram, Figure 4.2, is pressure versus theta in a non-firing or motoring engine. Several differences may be seen in the motoring versus the firing engine plots above.

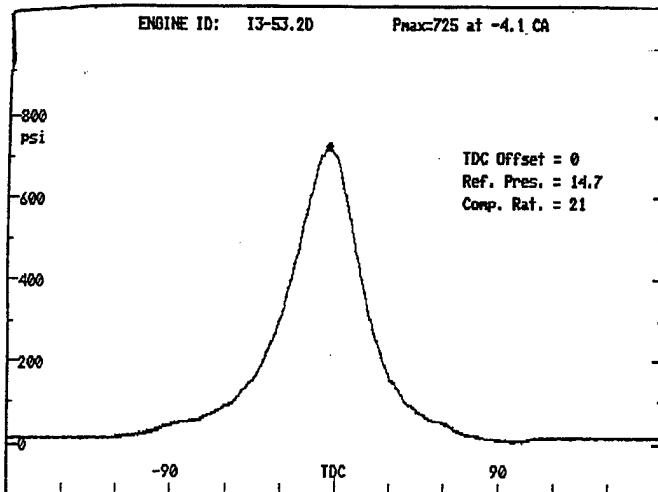


Figure 4.2 Pressure vs. Crank Angle Motoring Engine

The biggest difference is the maximum pressure reached. The motoring engine is basically acting as a compressor only. The maximum pressure is a direct result of the decrease in the volume of the cylinder. This pressure obviously increases as the piston moves upward, and decreases after TDC, the minimum volume, as the piston moves downward. Note that this curve is very smooth and that the increase in pressure starts at about minus sixty (-60) degrees.

The firing pressure curve, however, is different from the motoring curve. It also starts to increase at minus sixty degrees, but the curve reaches a much higher peak and is not as smooth as the motoring curve. The reason for the greater pressure is that fuel is added to the piston and burns to create gas and increased pressure within the cylinder. Fuel is injected into the cylinder a few degrees before TDC. In a diesel, fuel does not instantly burn as it does in a spark ignition engine. Some time is necessary for the fuel to mix with the air in the cylinder and for the fuel to heat up sufficiently to burn. Some combustion starts right at injection, but

the majority of the combustion occurs slightly later as the flame front of the combustion expands. This sudden increase in pressure is due to the fuel experiencing combustion.

The maximum pressure reached for all of these curves may not really be the exact pressure. If there is an error it only affects the magnitude, and not the shape nor the trends.

Another plot available through the software is the pressure verses volume curve. The software displays a plot of the pressure against the crank angle, however it really is plotting against the volume. This was verified with a MATLAB program and plot which converted the angle of the shaft to the volume in the cylinder. Plots of the ECA and MATLAB pressure-volume curve are Figures 4.3 and 4.4 respectively.

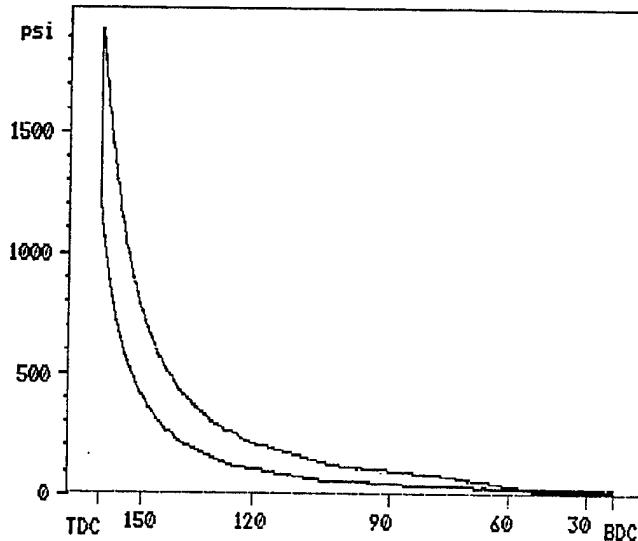


Figure 4.3 ECA Pressure - Volume Diagram (at 1800 RPM 125 ft-lb)

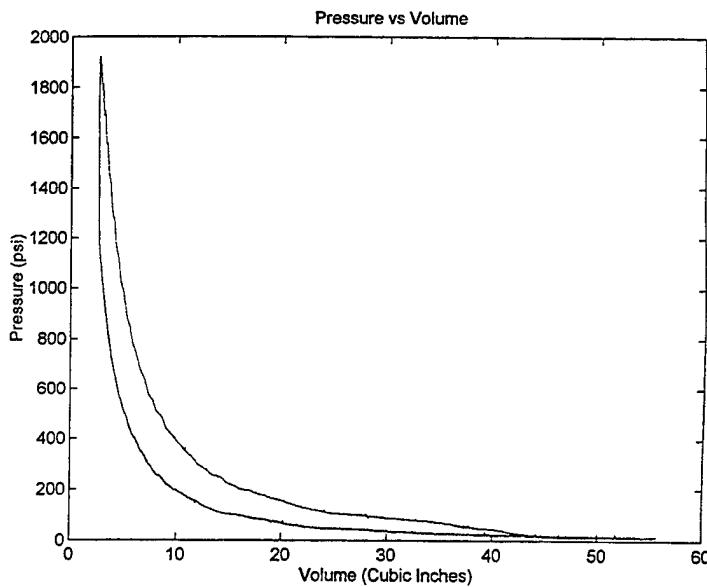


Figure 4.4 MATLAB Pressure - Volume Diagram (1800 RPM 125 ft-lb)

The shape of these curves is contrary to what was expected. They look like the curves of an Otto Cycle, gasoline or spark ignition, not like a Diesel Cycle.

The theoretical Otto cycle and Diesel cycle P-V diagrams are shown in Figure 4.5 and Figure 4.6. The Diesel cycle curves are supposed to increase as they do, but once combustion begins, expansion is supposed to be at a constant pressure.

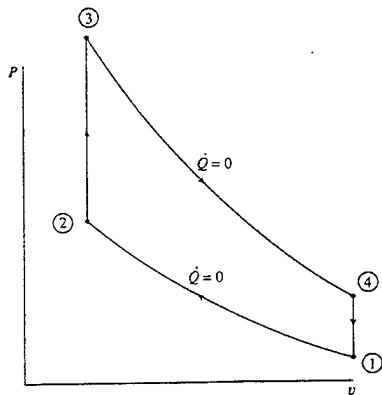


Figure 4.5 Theoretical Otto PV diagram [Ref 16]

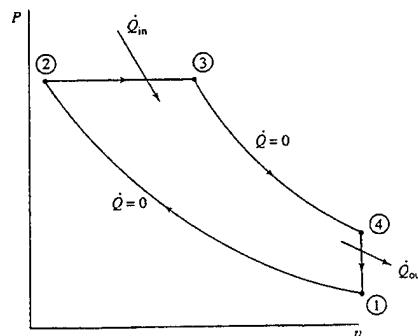


Figure 4.6 Theoretical Diesel PV diagram [Ref 16]

Another curve provided by the software is the log Pressure versus log Volume. This plot is very useful in computing the polytropic exponent. Pressure and volume relate in the following equation:

$$PV^n = \text{constant} \quad 4.1$$

The n is the polytropic exponent and for diesels and is typically between 1.34 and 1.37. [Ref. 15] In this plot, the n will be the slope of the line.

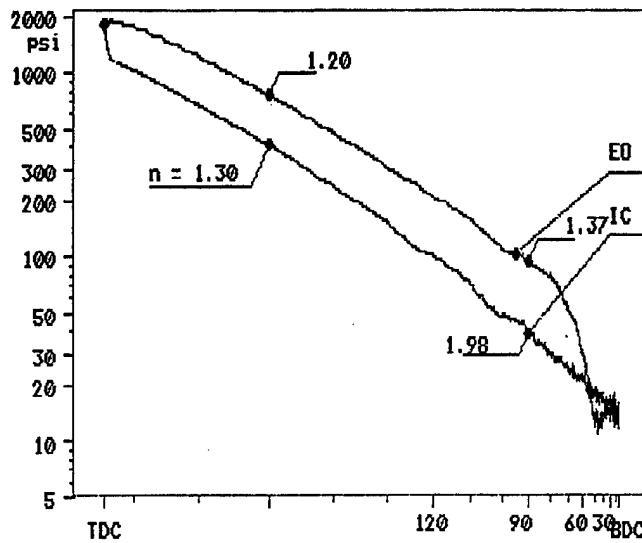


Figure 4.7 Log Pressure vs. Log Volume Plot at 1800 RPM and 125 ft-lb

The polytropic exponent for this system ranged between 2.0 and 1.2, yet predominately remained around 1.3. A polytropic system would theoretically follow the same straight line up and down with a slope of 1.31-1.37. The results are similar.

The last plot examined is an entropy versus temperature diagram. This plot is subject to many influences from inlet temperatures to reference pressures. The ECA does not take

any temperature measurements and computes this quantity. Reference 2 Section 3 discusses how these results are derived and computed. Figure 4.8 is the result of this plot.

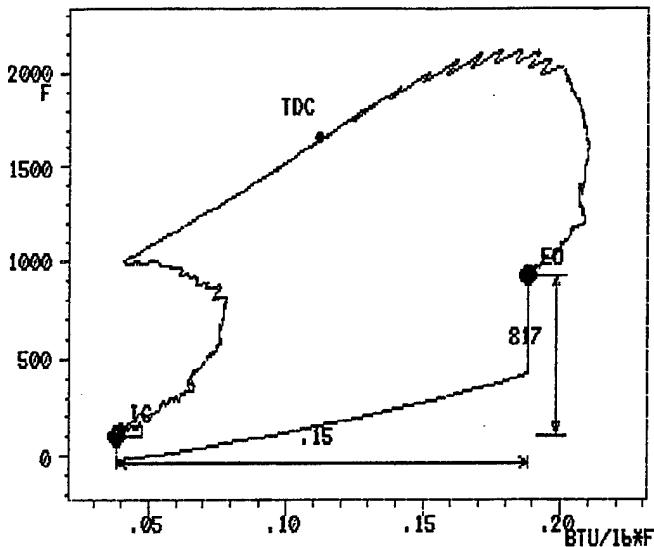


Figure 4.8 Temperature vs. Entropy

This plot resembles a theoretical T-S diagram for a diesel, but the actual quantities may not be correct. Engine at 1800 RPM and 125 ft-lbs.

4.2 ENGINE VIBRATION SIGNAL RESULTS

4.2.1 Signal Repeatability

The vibration signals from various points on the engine were analyzed. All vibration tests to be reported here were conducted at 1,800 RPM and 125 ft-lb torque. However, some other torques and speeds were tested and qualitatively compared. These conditions were chosen because it is about the middle of the engine's capabilities. This combination equates to about forty three (43) horsepower from an engine maximum of ninety three (93) horsepower. The torque is about seventy percent of the maximum.

Due to the variable nature of the diesel process, the requisite number of cycles for

adequate data and noise elimination had to be determined. One hundred individual cycles were taken. These cycles were then translated to a file which MATLAB could understand and each one compared. The cycles all resembled each other, but were not identical. Averages were performed on the cycles in groups of five. These signals also had similar attributes, but they did not look the same. Averages of twenty five and fifty were also examined. The best most repeatable signal was obtained when 100 cycles were taken. The 100 cycle signal eliminates the most noise without eliminating the entire signal. One hundred cycles also takes very little time to collect, so engine throttle drift would have little effect.

Figure 4.9 shows data collected at one time and the averages of a certain number of cycles.

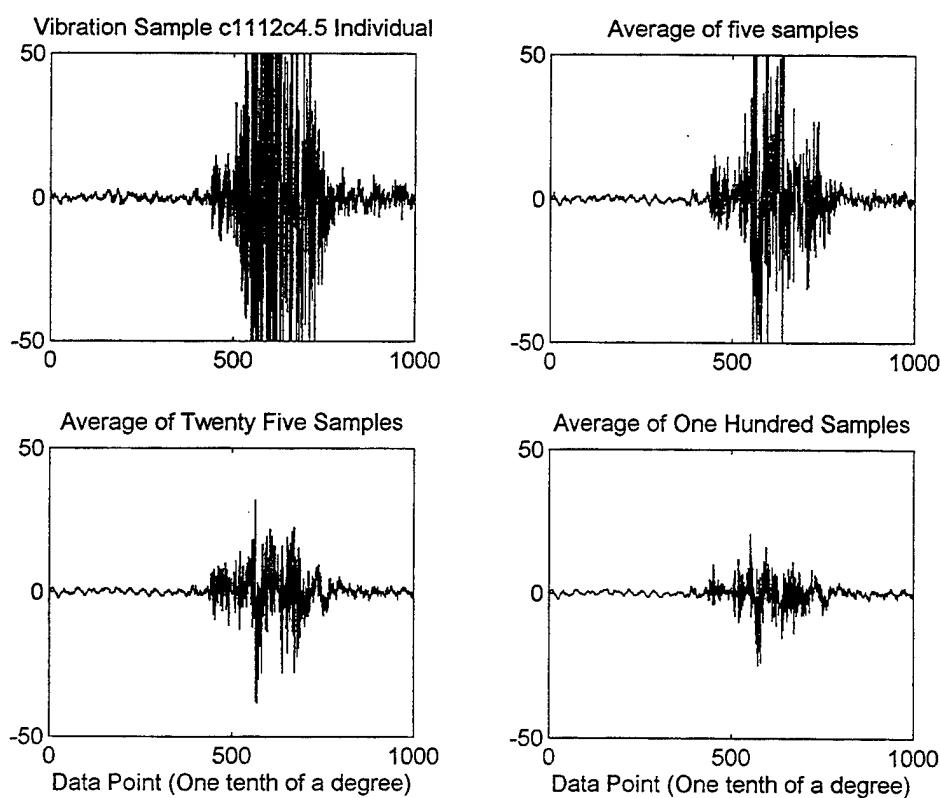


Figure 4.9 Comparing one cycle of data to the averages of 5, 25, and 100 cycles.

Figure 4.10 is a collection of four data samples taken at different times but over one hundred cycles.

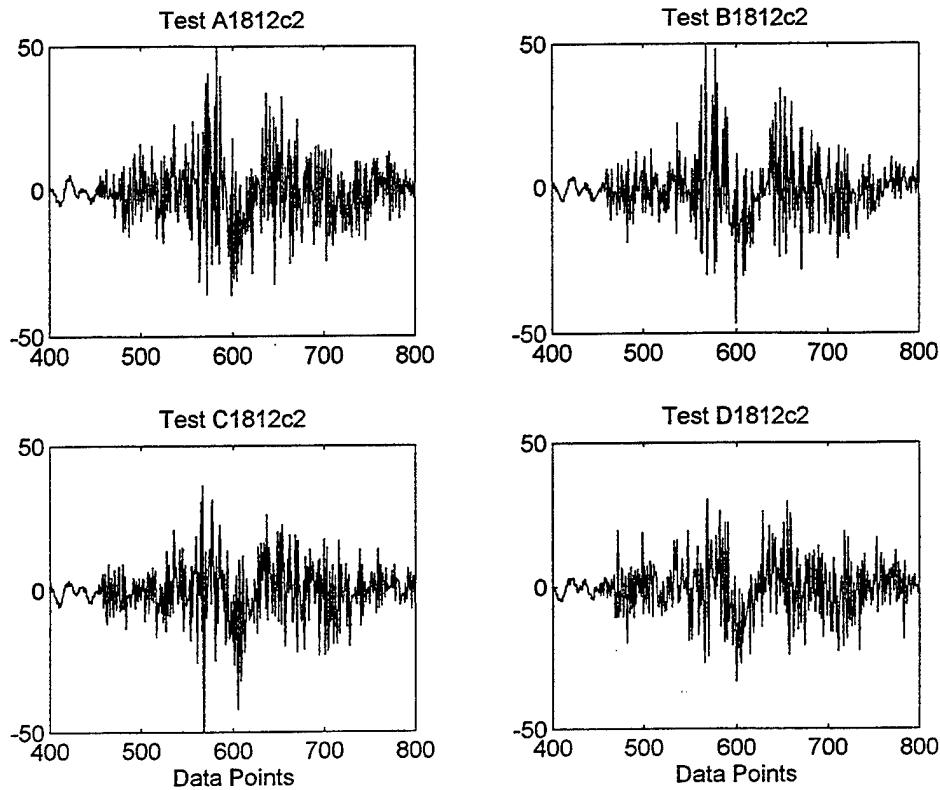


Figure 4.10 Four different samples of 100 cycles averaged.

An average of 200 cycles was also taken, and this signal looked virtually the same as the 100 cycle signal, however it was determined that the advantage of the smoother signal did not outweigh the extra time required, nor the chance for engine drift.

Another way to tell that 100 cycles is the correct number of cycles to use is to look at the Fast Fourier Transform (FFT) of the signal. The FFT takes a signal and breaks it down into its different frequency components. Figure 4.11a is the FFT of the average of 100 FFTs

of individual signals. Figure 4.11b is the FFT of 100 cycles averaged together. Figure 4.11c is the magnitude of the FFT with no random phase cancellation. The curves are identical.

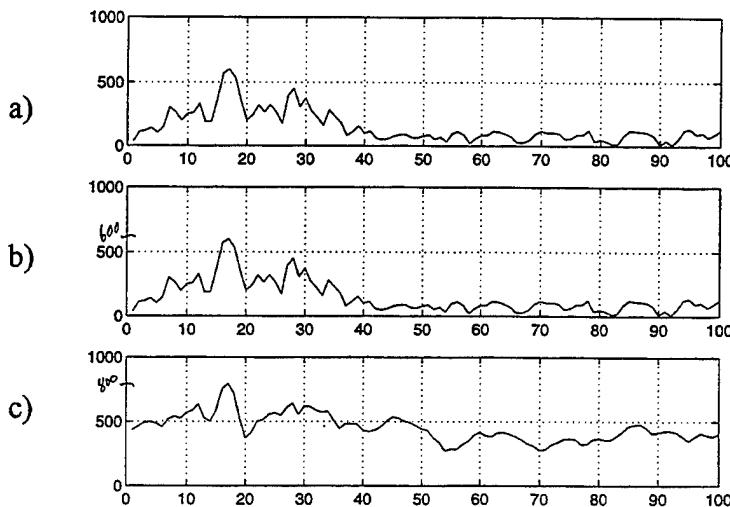
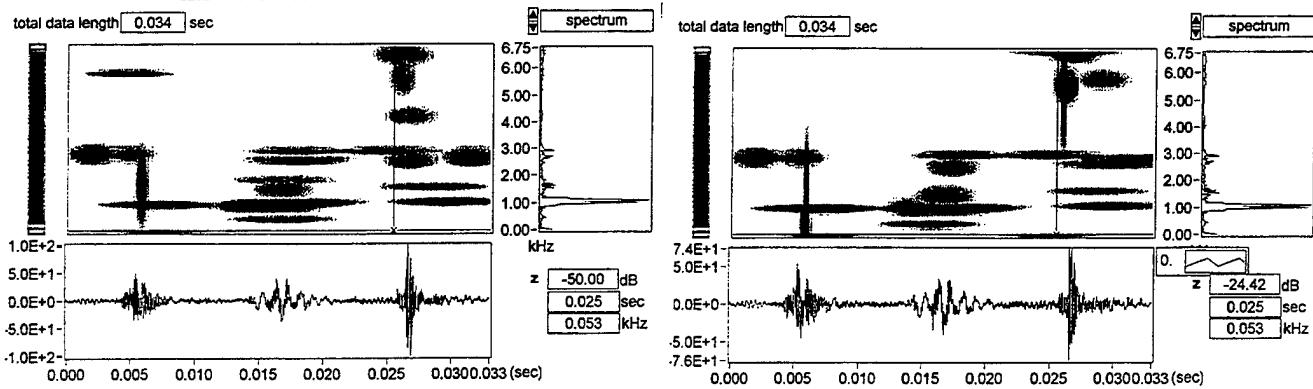


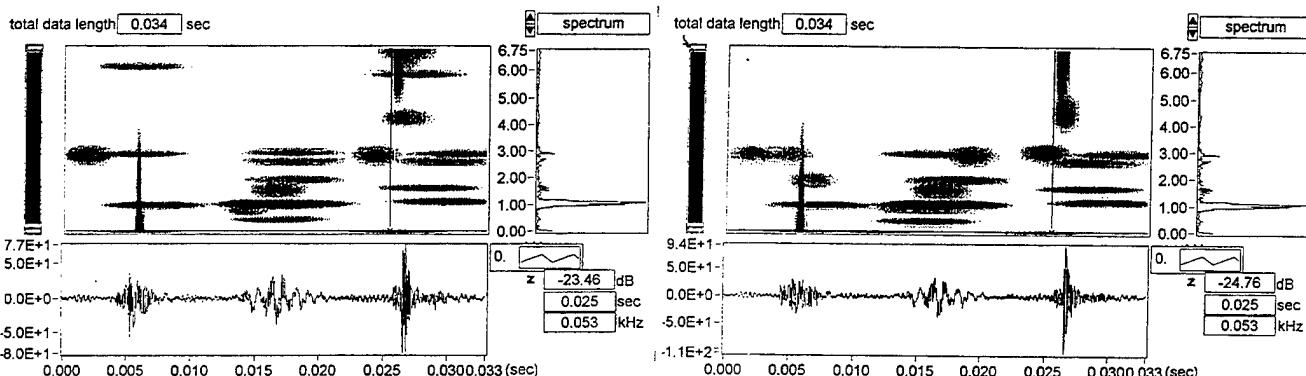
Figure 4.11 Fast Fourier Transforms of a) 100 signals then averaged together, b) 100 signals averaged, and c) the magnitude of FFT of 100 averaged together.

The vibration signals were averaged over 100 cycles to eliminate noise. This averaging creates a phase locked ensemble average of the data. The averaged signals obtained from all of the transducers were very repeatable both from one day to the next and from one hour to the next. They did not change much once the engine was at thermal equilibrium. The signals labeled A, B, C, D 1812C2.LOG are signals taken from a running engine at 1,800 RPM and 125 torque over the course of one hour. All of the signals are virtually the same, yet they were taken forty five minutes apart. The frequency components of each signal are the same and the JTFA islands are basically the same. These signals are also representative of signals from one day to the next.



A1812c2- 15 minutes run time

B1812c2- 30 minutes



C1812c2- 45 minutes

D1812c2- 60 minutes

Figure 4.12 Vibration Sensor C2 sampled over one hour. At 1,800 RPM and 125 ft-lb

The repeatability of a signal is a key point for using vibration signals as a method for CBM. If the signals did not repeat, they would be useless for trend analysis.

The vibration signal of the engine is due to some process in the engine occurring. The propagation of that wave through the engine may take many paths, but through out the engine there is a large damping factor. The firing cylinder in the cylinder is a very large force imparted over a very short time. It is very impulse like. The resulting signal damps out usually before the next cylinder firing.

4.2.2 Effects of Head Bolt Torque

Through experimentation, head bolt torques were varied from 160 ft-lb through 190 ft lb. This was done to prove that the engine would not be susceptible to minor variations in assembly. In the course of an engine's service life, the head may be removed for work on one component or another. If the tightening of these bolts is done at a different torque, will it completely change the vibrational properties of the engine? If a particular series of engine is being assembled with fundamentally the same parts, will one engine with the head bolt torque set to the correct torque range differ from and engine with a slightly different torque setting? Could this signal be used to diagnose potential problems?

The head bolts are 5/16 inch diameter bolts and are required to be torqued to 170-180 ft-lb [Ref. 14]. This condition equates to putting a preload force of approximately 20,000 lbs on the bolt. The preload equates to about 80% of the calculated proof load of a bolt. According to Shigly [Ref. 17], a reused bolted connection should be 75% of the proof load which is consistent. He also states that torque is a poor measure of the strain in a bolt or bolted connection due to the different variations in friction available. Shin concludes that little or no change in vibration response is observed for bolt preloads above 40 percent of their bolt proof load [Ref. 18]. The head bolts are clearly above 40 percent of their proof load, so no

observable shift should be observed in vibration response.

Data were collected from the engine in a baseline condition. In this baseline condition, five of the head bolts were tightened to 180 ft-lb torque. Two of the bolts were tightened to 170 ft-lb and the remaining bolt was at 190 ft-lb. Data were collected with the engine at 1800 RPM and 125 ft-lb torque. The head bolt torques were then changed to 160 ft-lb and 190 ft-lb torque and vibration data were collected with the engine at the same operating conditions as before.

The head bolts were loosened and tightened in accordance with the engine Service Manual [Ref. 14]. The bolts were tightened in the pattern listed in the figure below.

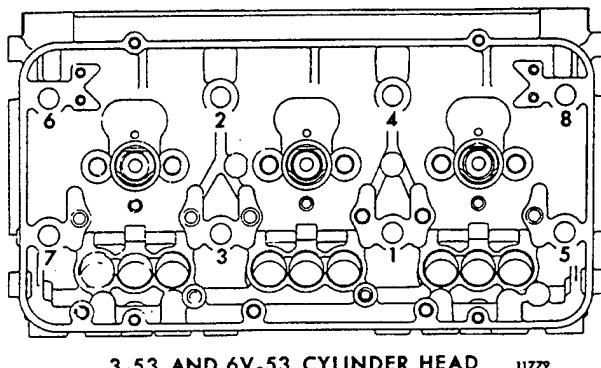
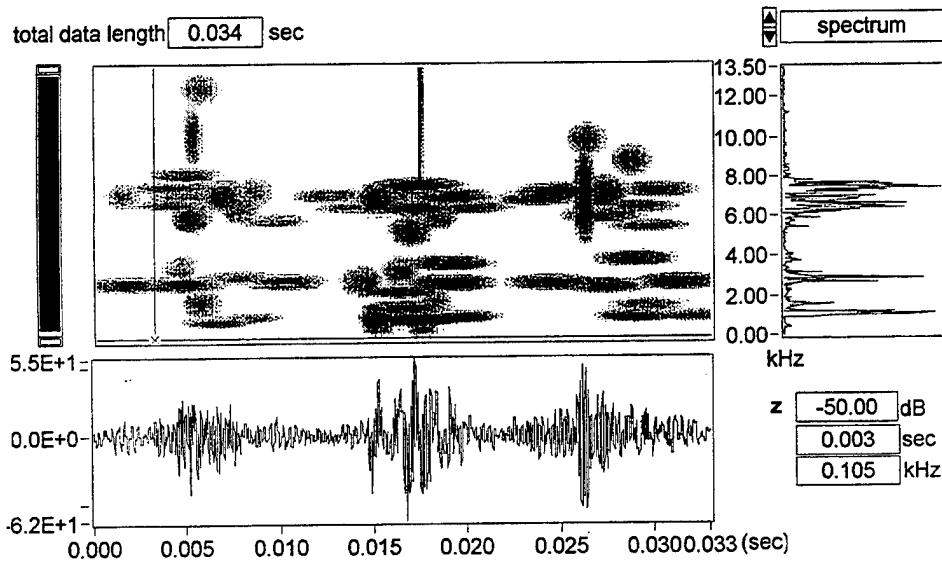


Figure 4.13 Head Bolt Tightening Pattern

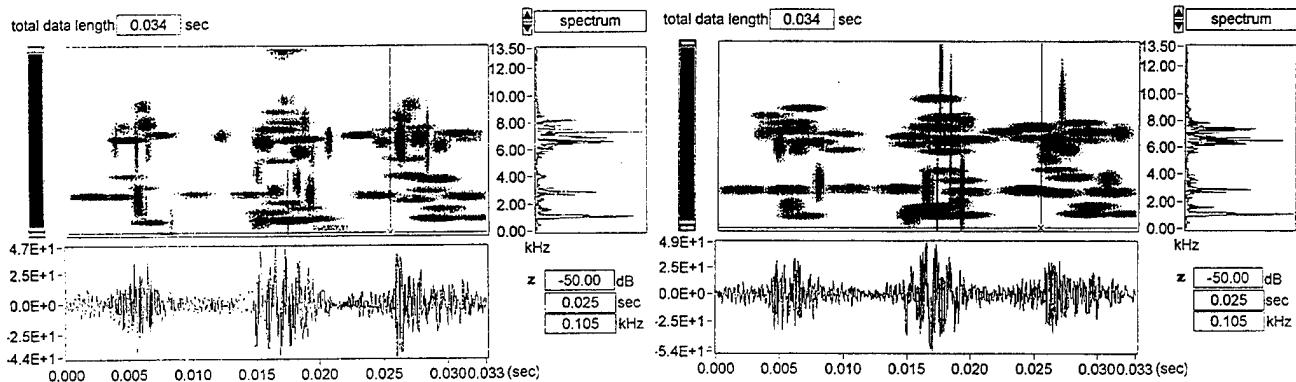
This pattern was repeated several times each time increasing the torque by fifty ft-lb until the desired torque was achieved. The torque wrench used was a Craftsman Digitork Wrench. This wrench is accurate to within three percent of the set torque.

Analysis of these data revealed that the torque had no measurable effect on the vibration of the head bolts or on the sensors placed on the engine, head assembly or the

engine base. This may be observed in Figure 4.14.



a) Engine Cylinder Head Baseline data 180 ft-lb torque.



b) Engine Cylinder Head 160 ft-lb torque

c) Engine Cylinder Head 190 ft-lb torque

Figure 4.14 Cylinder Head Assembly Vibration Data with Varying Bolt Torques

This is a very important point because it eliminates the variation that may result from bolts loosening or a less than careful mechanic tightening the head bolts.

4.2.3 Frequency Components of Signals

The results revealed three major frequencies excited while the engine was running. The frequencies were predominantly independent of engine speed and torque. The frequencies excited were about 1,100 Hz, 3,000 Hz, and 7,000- 8,000 Hz. It varied slightly from run to run, but those were the predominate peaks.

To explain the reasons for the different frequencies, we may model the engine head assembly as a simple mass damper system mounted on eight springs. This may be seen in Figure 4.15.

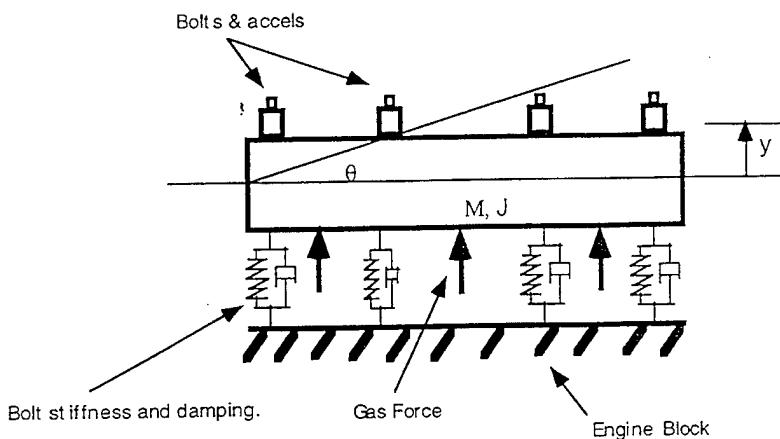


Figure 4.15 Mass Damper Model

The model equates the head assembly as a block and the eight head bolts as springs. The fundamental frequency of this system should follow the following equation:

$$f = \frac{1}{2\pi} \sqrt{\frac{K}{m}} \quad 4.2$$

The variable m is the mass of the system, which for this model is head assembly mass. It was measured to be 93 lb or 42 kg. The variable K is the stiffness of the springs and is

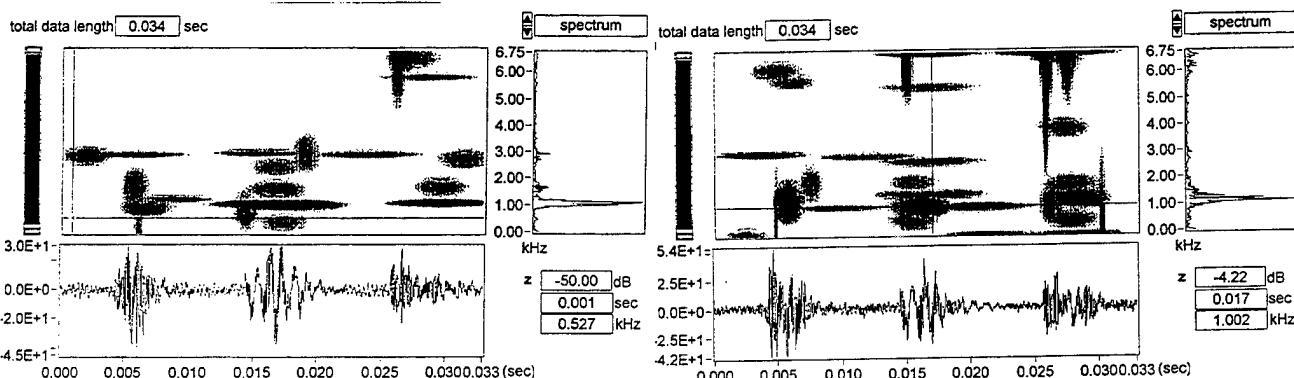
$$K = \frac{AE}{l} \quad 4.3$$

The area of the spring (A) is the cross sectional area of the spring. For the purposes of this model, it may be defined as eight (the number of head bolts) times the cross sectional area of one bolt. The diameter of a bolt is 9/16 inches or 0.01428 m. The free length of the bolt (l) is 4.75 inches or 0.1207 m. The E of the steel bolts is modulus of elasticity which is about 200 GPa. This yields a K of $2.125 \times 10^9 \text{ kg/sec}^2$, which yields a frequency of 1,132 Hz. This is very close to the 1,100 Hz frequency shown in the plots.

In reality, the bolts are not simple springs and the bolted connections between the head and the cylinder block assembly may need to be refined. According to Shigley, the stiffness of the bolt is a function of the entire length of the bolt, the free length plus the threaded portion [Ref. 17]. This results in a K of $1.346 \times 10^9 \text{ kg/sec}^2$. This is a difference of thirty six (36) percent and yields a frequency of about 900 Hz for the entire length of the bolt. He also states that the stiffness is a function of the bolt and the members being connected [Ref. 17]. The members and the bolt act as a single member. Calculating the stiffness with this method results in a stiffness K of $2.00 \times 10^9 \text{ kg/sec}^2$ and a frequency of 1,098 Hz.

All of these techniques yield results of the same order of magnitude as the observed 1,100 Hz signal. Thus we conclude that the observed 1,100 Hz is due to the natural frequency of the head bolt and its connections.

The frequencies had different regions when they were most strong. The 1,100 Hz frequency was most prevalent during the combustion in the cylinders. This may be explained by the force of the combustion straining the bolts and exciting them. The 1,100 Hz signal does not appear in the regions between the explosions in each of the cylinders. To further emphasize that 1,100 Hz is due to the bolts is that it is the only frequency in some bolt head transducers and the largest frequency in others as shown in Figure 4.16.



a) Cylinder Head bolts C2

b) Head bolt C5

Figure 4.16 1,100 Hz Frequencies in Head bolts at 1800 RPM and 125 ft-lb

The other frequencies are believed to be acoustical vibration in the cast iron cylinder block. The speed of sound in solid is

$$C_s = \sqrt{\frac{E}{\rho}}$$

4.4

The modulus of elasticity, E, for cast iron is about 90 GPa and it's density, ρ , is 7,000 kg/m³. This yields a speed of sound of 3,500 m/s in cast iron. The length of the engine block is about eighteen inches, of 0.457 m. The height is twenty four inches, 0.61 m, and the width ten inches, 0.254 m. These measurements are approximate because of the varying dimensions of the engine. The outer casing of the engine is made of aluminum. The service manual indicates that the engine cylinder block is made of cast iron. The case may be that the engine block is cast iron, but the only the outer casing is aluminum. The actual dimensions of the cast iron parts may vary from our predictions. Velocity is distance divided by time, therefore the inverse of time, frequency is velocity divided by distance.

For a sound wave to travel back and forth through a solid block as a standing wave, the length of the wave must be roughly equal to the length of the block or half of the wavelength must be equal to the block length. Using this information the following results are revealed:

Engine Dimension	Distance	Frequency	One Half Frequency
Length	0.457 m	7841 Hz	3920 Hz
Height	0.610 m	14114 Hz	7057 Hz
Width	0.254 m	5877 Hz	2938 Hz

Table 4.1 Expected Acoustical Frequency due to a Standing Wave.

The half frequencies for the height and the width both approximate two of the frequencies observed. The vibration transducers were measuring in the Y direction, up and down, so they are probably most sensitive to this vibration.

The transducers mounted on the engine measured more frequencies than the head bolt mounted transducers. The 3,000 Hz signal is fairly prevalent throughout the entire cycle. It

is strongest during the combustion times, but it does not always go away. Similarly, the 8,000 Hz signal does not really go away throughout the entire cycle. This indicates that there is standing wave in the engine block which is not damped out quickly as seen in Figure 4.17.

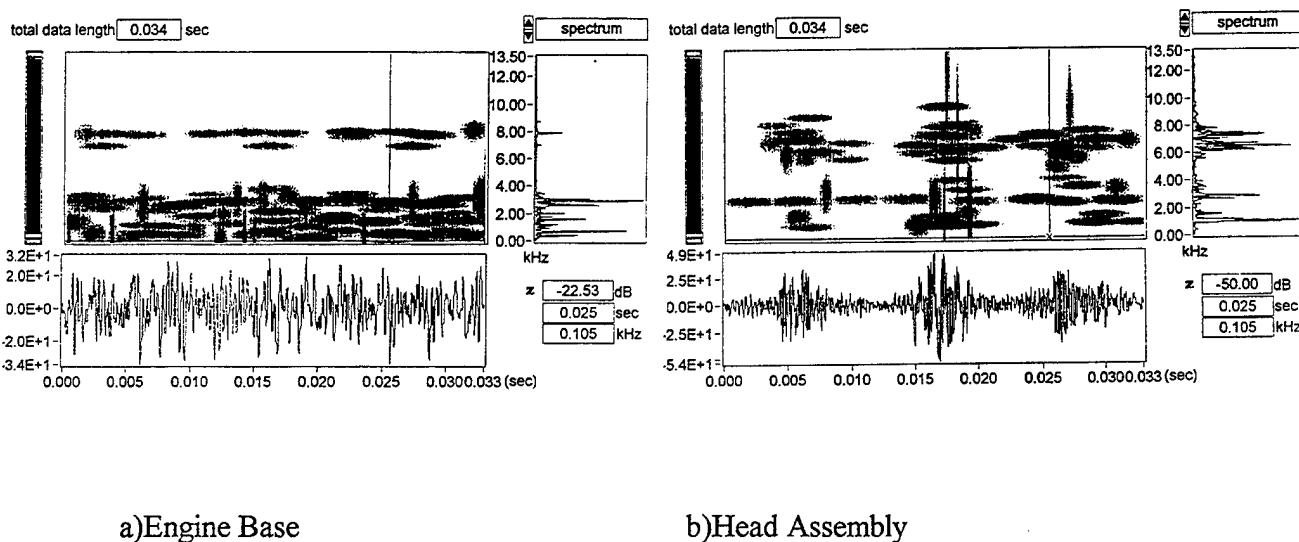


Figure 4.17- 3,000 Hz and 8,000 Hz Frequency in Engine Assemblies.

4.3 DIAGNOSIS THROUGH SHAFT SPEED VARIATION

A Hewlett Packard 53310A Modulation Domain Analyzer was used to measure the time interval between encoder pulses. This device measures and records the time difference between pulses of a repeating signal. This device is normally used for radar pulses, however it has an interesting application for engine research. This analyzer was used in conjunction with the optical encoder installed on the engine as a possible diagnosis tool. This research was begun without the knowledge of the research in this field reviewed earlier.

As alluded earlier, an engine cycle is considered one complete revolution of the crankshaft for a two stroke engine. In a revolution for this engine all three cylinders fire. One could hypothesize that the rotation of the engine will not be completely constant. There may be periods of acceleration of the crankshaft and periods of deceleration. The acceleration times should be just after the combustion in a cylinder, and the deceleration due to compression just before combustion begins in the next cylinder. This leading then lagging the average rotational speed may be a predictor for engine health.

The Modulation Domain Analyzer (MDA) uses neither sampling time nor frequency to gather it's data; it times events between some threshold value. In other words, it starts and stops counting only when the desired event has taken place. Therefore, sampling frequency and sampling rate are immaterial. The only source of error for this experimental setup would be the alignment of the optical encoder and the vibrations of the mount. If it were not exactly aligned a slight deviation in signal would be expected due to the torsional vibration. There may also be some error associated with an irregular distance between the graduations of the glass wheel of the encoder.

The optical encoder installed on the engine provides an output signal which is a five volt (5V) square wave. The signal is sent as the engine turns through every tenth of a degree. This signal was timed by the MDA and plotted. A sample result of this is shown in Figure 4.18.

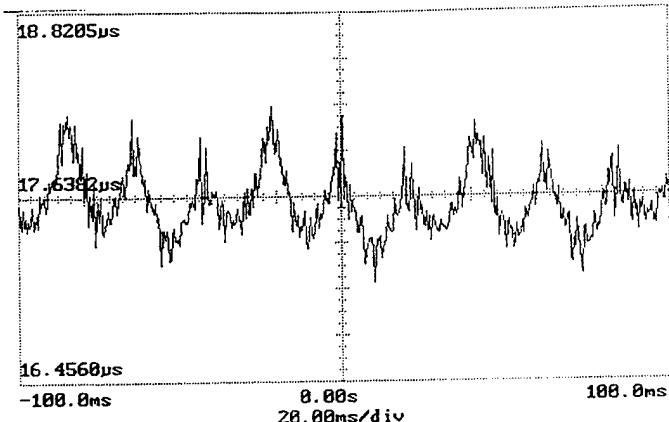


Figure 4.18 Shaft Speed Variation 950 RPM 144 lb-ft

The plot's central line is the average time between signals of the encoder. The RPM of the engine may be computed from

$$RPM = \frac{60\text{sec/min}}{time(\text{sec}) * 3600\text{counts/rev}} \quad 4.5$$

The peaks above the central line are decelerations of the engine. The time listed in the top left hand side of the box is a reference for the amount of this deceleration. The peaks are decelerations since more time required for the signal to be received. Similarly, the dips below the central line are the acceleration of the shaft speed. This variation was approximately three percent (3%) or the average encoder speed.

Some plots appear to be substantially below or above this line. This is due to the engine speed drifting slightly during data acquisition. This drift may be corrected by recycling the Autoscale function of the MDA.

The firing of each cylinder is clearly visible in figure 4.18. The cylinder patterns is most clear when the engine was operating at low RPM and high torque. This was as expected

as explained by Mauer [Ref. 6]. With low RPM and high torque, the variation between cylinder pressures is greater than the overall rotational inertia of the system. This is not the case when the RPM is increased and the torque lessened. When there is a greater RPM and less torque the rotational inertia of the system is far greater than the variation between the cylinders. This result is evident in Figure 4.19.

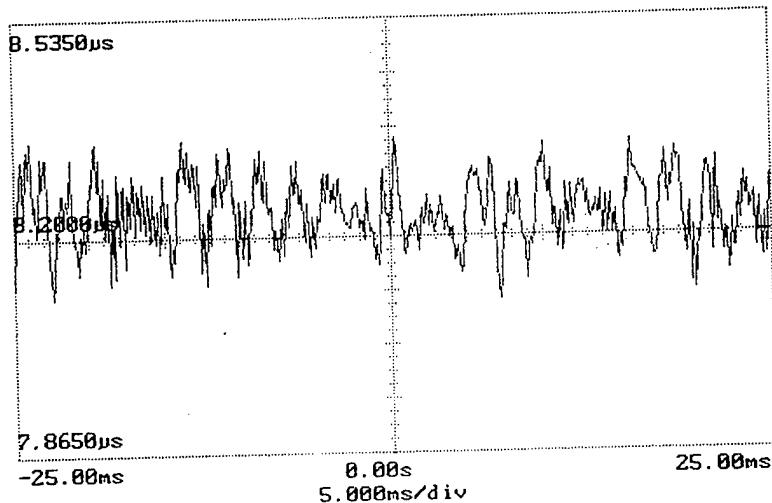


Figure 4.19 Shaft Speed Variation 2000 RPM 50 lb-ft

There were some problems which could not be resolved with the MDA in the short time which we had it. The MDA measures the difference between the optical encoder signals, but it may also be referenced to another signal to assure that the individual cylinders may be identified. We set this reference as the TDC signal from the optical encoder. This should have referenced the plot with cylinder one in the center. Sometimes this was the case, yet most times the plot would “roll” and we could not determine exactly which cylinder was which. This ability to differentiate between cylinders is essential for engine diagnostics.

An attempt was made to simulate low pressure in a cylinder by loosening the glow plug in number two cylinder. The simulation was not successful due to fearing permanently

damaging the threads in the glow plug hole and creating a projectile of the glow plug. There was not sufficient leakage to affect the engine nor to be detected by the MDA. There was not sufficient time to investigate this safely or efficiently.

A collection of more shaft speed variation plots are included in Appendix F.

V. CONCLUSIONS AND RECOMMENDATIONS

5.1 SUMMARY

The diesel engine was instrumented with high speed sensors and all of these sensors appear to be fully functional. Various methods for engine diagnosis were investigated. The data gathered leads to the following conclusions: The statistical nature of the signals was established and one hundred cycles appear to be a sufficient number of cycles to eliminate noise but not eliminate too much of the base signals. Tests were conducted to assess the impact of cylinder head bolt torque and it was determined to have no measurable effect on the vibrational signals of the engine. The predominate source of measured vibrations appears to be the natural frequency of the bolts and reflected acoustic waves within the engine. A possible method of diagnosis was investigated using shaft speed variation and seems to be most effective in which have low speed and high torque. Safe operating procedures were established for the engine and it's auxiliary systems.

5.2 ENGINE CYCLE ANALYZER

The Engine Cycle Analyzer is primarily an instrument for engine development. The conclusions from ECA data are the following: The pressure vs. volume curve looks more like an Otto cycle than a Diesel cycle. The pressure vs. theta curves reveal beginning and end points of combustion where motoring curves are smooth. Finally, the entropy vs. temperature plot reveals a negative change in entropy which cannot be accounted for.

5.3 VIBRATIONAL ANALYSIS

The Joint Time Frequency vibrational analysis of the engine has the following

conclusions: The vibrations were due to combustion and the ringing that it causes. These signals appeared to be specific to each cylinder, because the signals from one sensor to the next subjectively looked to be the same for a specific cylinder. The engine is relatively symmetric and a geometrical similarity of signals was expected, but this did not appear to be the case. Signals had very different characteristics depending on sensor location. The vibrations are believed to be due to standing acoustic waves in the engine block and head assembly and due to the natural frequency of the head bolts. The signals from the bolts ring down very quickly, but the acoustic waves do not seem to ring down. The signals appear to be independent of both speed and torque. One hundred engine cycles of data averaged together seems to be the least number of cycles required consistent for signal repeatability. The signals in between the cylinder explosions are extremely repeatable and may be signals from the engine such as gear teeth, blower, or pumps.

5.4 SHAFT SPEED VARIATION

The variation of shaft speed holds promise in detecting cylinder firing problems. It gives the best results under low speed and high torque. This method is the least invasive of the techniques and does not really change any parameters of the engine. Another optical encoder with lower resolution, perhaps 360 counts, may be better since the higher frequencies do not appear to be of interest and more cycles can be obtained.

5.5 RECOMMENDATIONS

Some additional instrumentation may be interesting for the investigation of this engine. Installation of a turbine meter in the exhaust system for flow and temperature readings may be of interest. Similarly installing an *Orsat gas analysis apparatus* for the volumetric

percentages of the exhaust gas may also be of interest for both this investigation and the thermodynamic study of diesel engines. A pressure and temperature sensor should be obtained to measure the pressure and temperature in the air box manifold. This would become the reference pressure for the piezoelectric pressure transducers. This new pressure transducer may be placed in the spark plug mounting hole in the side of the engine.

The calibration of the inlet air flow turbine needs to be checked.

A motoring electric dynamometer would allow exact control of the torque on the engine enabling more constant conditions and would be useful in this type of work.

A permanent twelve volt electric power supply system should be installed on the test stand which would provide power for the dynamometer during operations and eliminate the constant drain on the batteries.

The ECA is somewhat limited by its ability to only gather four channels of data plus the optical encoder. Another device that could handle more channels of information would be better.

Further research should change the injectors to see if the individual cylinder signal is due to the injector or the cylinder itself.

A torsional model of the engine and flywheel system should be developed based on Holtzer's Transfer Matrix.

A new more rigid mounting for the optical encoder may be necessary. This mounting should be able to accommodate a lower resolution optical encoder and should also be designed for torsional stability to eliminate the possibility of the encoder introducing errors in the shaft speed variation investigation.

Another shaft speed encoder should also be mounted on the flywheel end of the engine. This encoder could operate on the teeth on the flywheel, there are approximately 132 teeth, or it could be mounted in the transmission housing between the flywheel and the water brake. Comparison of these two signals should also be of interest.

The results of the encoders could then be compared with the pressure data. Experiments could be performed by inducing faults in a certain cylinder such as the following: the fuel metering rack misadjusted for that cylinder, using a hollowed glow plug to vent combustion gases as might happen with leaking rings or bad exhaust nozzles, or the spray tip holes on the fuel injector may be bored out or plugged to simulate a worn or clogged injector.

Several support documents should be obtained from Detroit Diesel to aid in engine research. For use in the vibration investigation, detailed drawing of the engine should be obtained so that the number of teeth for each gear may be examined, the exact size of all the parts may be determined. Pressure-volume data should also be obtained under several different load and RPM conditions.

A method for turning the engine over by hand must also be devised. Perhaps a ratchet system that would fit around the optical encoder mounting onto the crankshaft pulley would be good. Another possible location for a turning system is the drive shaft between the flywheel and the water brake.

APPENDIX A

MASTER LIGHT OFF PROCEDURE (MLOP) DIESEL CELL

SYSTEM VERIFICATION ALIGNMENT PROCEDURES AND OPERATING PROCEDURES

PLACING THE DIESEL TEST CELL INTO OPERATION

1. Conduct visual inspection of diesel test cell and verify the following:
 - a. Ensure all drip pans, piping trenches, and the deck are free of oil, fuel, or any flammable liquids.
 - b. Ensure all flammable liquids are stored properly in the flammable liquids locker.
 - c. Ensure all small parts, equipment or, tools are properly stored.
 - d. Verify that the diesel test cell and work area fire extinguishers are fully charged.
 - e. Inspect all piping runs and accessories for loose connections, damage, or leaks.
 - f. Ensure all valve hand wheels are installed and valve labels are in place.
 - g. Inspect the air intake louvers for blockage.
 - h. Check the engine battery voltage. Place the batteries on charge if voltage is below 12 VOLTS.
 - i. Check fuel level in fuel oil storage tank. Ensure enough fuel is present to support diesel operations.
 - j. Verify that the water storage tank is full.
 - k. Verify that all glow plugs, pressure transducers, or cylinder plugs are installed.
 - l. Inspect the optical encoder. Ensure the mounting is secure and no cabling will become entangled.
 - m. Inspect vibration transducer wiring. Ensure it is clear of all rotating parts and

connected as required

2. If the diesel test cell and the gas turbine test cell have been idle for more than 30 days, the water system and fuel system must be recirculated prior to placing the systems into operation.
3. If the ambient air temperature is less than 50°F, the fuel system must be recirculated in order to ensure no paraffin separation is present.
4. Ensure the cooling water system filter is clean and free of excessive particulate.

SYSTEMS ALIGNMENT

Note: All steps followed by an asterisk (*) are located outside on the pump pad. All others are considered to be in the test cell.

WATER SYSTEM ALIGNMENT

1. Ensure the following valves are in the fully open position:
 - a. Water storage tank suction valve CW-1*
 - b. Water supply pump suction valve CW-2D*
 - c. Water supply pump discharge valve CW-4D*
 - d. Heat exchanger inlet valve CW-10*
 - e. Heat exchanger discharge valve CW-12*
 - f. Dynamometer sump tank supply valve CW-6D
 - g. Return pump discharge valve CW-9D
2. Ensure the following valves are in the full closed position:
 - a. Gas turbine supply pump suction valve CW-2GT*
 - b. Gas turbine supply pump discharge valve CW-4GT*
 - c. Gas turbine to diesel cross connect valve CW-5*

- d. Heat exchanger bypass valve CW-11*
- e. Gas turbine return pump discharge valve CW-9GT
- f. Dynamometer unloader bypass valve CW-7D
- 3. Place the local heat exchanger breaker in the AUTO position.*
- 4. Place the local cooling water supply pump breaker [P-IN5] in the AUTO position.*
- 5. Place the local cooling water return pump breaker in the AUTO position.*

FUEL OIL SYSTEM ALIGNMENT

- 1. Ensure the following valves are in the fully open position:
 - a. Fuel oil supply pump suction valve FOS-2D*
 - b. Fuel oil supply pump discharge valve FOS-5D*
 - c. Fuel oil supply cell isolation valve FOS-6D
 - d. Fuel oil return valve FOS-12D
- 2. Place the local fuel supply pump breaker [P-IN7] in the AUTO position.*
- 3. Ensure the following valves are in the fully closed position:
 - a. Gas turbine fuel oil supply pump suction valve FOS-2GT*
 - b. Fuel oil supply cross-connect valve FOS-3*
 - c. Gas turbine fuel oil return valve FOS-10GT*
- 4. Ensure the dynamometer to engine fuel line quick disconnect is properly connected.
(Under test stand)

AIR SYSTEM ALIGNMENT

- 1. Place the remote LOUVER switch to the OPEN position.

2. Ensure air flow turbine meters are connected to the dynamometer instrumentation rack.
3. Ensure that the AIR FLAPPER (for Emergency Kill) is open and latched.

OIL SYSTEM VERIFICATION

1. Verify the lubrication oil level in the oil sump is above the low mark on the dipstick.
2. Ensure the oil pressure sensing line is connected to the dynamometer instrumentation rack connection.

COMPUTER AND DATA COLLECTION EQUIPMENT

1. Determine what data is required for collection.
2. Utilizing the sensor map (Appendix B) connect desired sensors to the four channels available and record these connections.
3. Turn on the following equipment and allow at least twenty minutes of warmup time before data collection to allow thermal equilibrium.
 - a. Kistler 5010 Charge Amplifiers.
 - b. PCB Conditioning Amplifiers
 - c. Krohn-Hite Low Pass Filter
 - d. HP Oscilloscope
 - e. ECA Sensor Interface
 - f. Dynamometer Control Computer
 1. C:\>cd sf901
 2. C:\SF901>sf901
 - g. ECA Control Computer

ENGINE CHECKS AND ADJUSTMENTS

1. Ensure the battery charger is disconnected from the storage batteries.

2. Obtain the latest barometric reading.
3. Verify the IGNITION switch and the FUEL PUMP switch located on the control console are in the OFF position. (Down Position)
4. Place the LOAD CONTROL and THROTTLE CONTROL switches to the MANUAL position.
5. Ensure the LOAD CONTROL knob is set to minimum.
6. Ensure the THROTTLE CONTROL knob is set to minimum.
7. Verify the THROTTLE CONTROLLING RPM switch is in the THROTTLE CONTROLLING RPM position. (This switch is located inside the control console.)
8. Press the POWER ON push-button to energize the dyno control console.
9. Set the shaft OVERSPEED knob at 2800 RPM.
10. Set the TORQUE/POWER display knob to the LOW scale.
11. Set the SPEED display knob to the LOW speed scale.
12. Set fuel specific gravity. Either enter the proper setting through the SPEC. GRAVITY knob or input the fuel specific gravity into the SF-901 computer monitoring system in accordance with factory technical manuals. (SG=0.834)
13. Turn the FUEL mode knob to the A configuration.
14. Set the AIR-FUEL meter knob in the FUEL configuration.
15. Set the SENSOR knob to the 6.5" position.
16. Set the DISPLAY knob to the TORQUE position.
17. Set the control console switch to the TENTHS scale. (Under console on right)
18. Adjust the TORQUE ADJUST knob to zero. (Knob under console on left)
19. Set TEMPERATURE/METER SELECTOR switch to LOAD. Turn up LOAD knob to maximum and ensure indicator responds. Return LOAD setting to minimum.
20. Set TEMPERATURE SELECTOR switch to THROTTLE. Turn up THROTTLE

knob and ensure indicator responds. Look at the blue throttle control drum on the dyno end of the test stand and ensure that it responds accordingly. Return THROTTLE setting to minimum.

21. Set the WATER VAPOR PRESSURE knob to the correct setting. (Sling psychrometer is located under the barometer near the gas turbine control stand.)

ENGINE STARTING PROCEDURE

1. Depress the remote COOLING WATER RETURN PUMP start push-button; verify the MOTOR RUN light illuminates.
2. Depress the remote COOLING WATER SUPPLY PUMP start push-button; verify the MOTOR RUN light illuminates.
3. Depress the remote FUEL OIL PUMP start push-button; verify the MOTOR RUN light illuminates.
4. Depress the remote HEAT EXCHANGER start push-button; verify the MOTOR RUN light illuminates.
5. Turn the SHUTTER MOTOR switch to the OPEN position. Ensure that the intake shutters move to the open position.
6. Open DYNO BOOST valve on the dynamometer.
7. Ensure both test cell entrance doors are closed and latched.
8. Ensure the Gas Turbine Test Cell dyno sumps are not flooding.
9. Turn the IGNITION switch to the ON position. (Up position)

***WARNING:** The IGNITION switch must be on for at least 10 seconds prior to engaging starter to ensure the dynamometer has sufficient priming water.*

10. Turn the FUEL PUMP switch to the ON position. (Up position)
11. Set TEMPERATURE SELECTOR switch to throttle. Turn up THROTTLE knob to 8 volts on the volt-meter (approx. 40 %).
12. Depress and hold the STARTER push-button for 2-5 seconds.
13. Verify positive lubrication oil pressure on lube oil pressure gage.

14. Monitor all console warning lights.

WARNING: Abort start by releasing START button and pulling the black Engine Idle Cutoff T-handle, or the red EMERGENCY KILL T-handle.

- a. Time from STARTER push-button depressed to start exceeds 10 seconds.
- b. No engine oil pressure is indicated on the control console.
- c. A WATER SUPPLY warning light illuminates which indicates a supply water pressure of less than 15 PSIG is available to the power absorber.
- d. A DYNO PRIME warning light illuminates indicating the power absorber must be reprimed.
- e. A FUEL PRESSURE warning light illuminates indicating less than 4 PSIG fuel pressure.
- f. An unusual sound or vibration occurs.
- g. A fuel or lubrication oil leak is observed.

15. Adjust the throttle servo RPM control and the load torque control to match the engine operating condition.

16. Watch the servo motor on the engine stand while switching the THROTTLE CONTROL to SERVO. Ensure that the throttle is responding correctly before continuing.

NOTE: Always shift the switch controlling speed first then shift the one controlling load.

17. Switch the LOAD CONTROL to SERVO.

18. Increase RPM to 1300 and torque to 100 ft-lbs. Allow engine to warm up.

SECURING THE ENGINE

1. When finished operating, decrease the speed and torque set points to idle. Torque to 000. Throttle to 900 RPM.
2. Ensure Manual TORQUE knob is set at minimum.

3. Set TEMPERATURE SELECTOR switch to throttle.
4. Switch LOAD CONTROL and THROTTLE CONTROL to MANUAL.
5. Adjust THROTTLE CONTROL to idle (900 RPM).
6. Allow engine to cool down. Watch water jacket temperature and when it has decreased sufficiently secure engine by pulling the IDLE SHUTDOWN T-Handle.
7. Ensure engine stops. If it does not stop pull the red EMERGENCY KILL T- Handle.
8. Switch the IGNITION and FUEL PUMP switches to OFF.
9. Depress the remote COOLING WATER SUPPLY PUMP stop push-button; verify the MOTOR RUN light extinguishes.
10. Depress the remote COOLING WATER RETURN PUMP stop push-button; verify the MOTOR RUN light extinguishes.
11. Depress the remote FUEL OIL PUMP stop push-button; verify the MOTOR RUN light extinguishes.
12. Depress the remote HEAT EXCHANGER stop push-button; verify the MOTOR RUN light extinguishes.
13. Close the DYNO BOOST valve.
14. Close the Water Supply and Return valves.
15. Secure power to computers, data collection equipment and the test console as required.
16. After cell has cooled, close vent louvers.

APPENDIX B
DATA ACQUISITION SUMMARY

Date _____

Accelerometers on Engine: (Normally will be cables and channels 2-4)
 Locations see Over:

Number	Plane	Data Cable (1-4)	PCB Channel (Chg Amp)	SF-891 Channel ECA Interface	Oscilloscope Channel
Head Bolts					
1	X	_____	_____	_____	_____
2	Y	_____	_____	_____	_____
3	Z	_____	_____	_____	_____
4	Y	_____	_____	_____	_____
5	Y	_____	_____	_____	_____
6	Y	_____	_____	_____	_____
7	Y	_____	_____	_____	_____
8	Y	_____	_____	_____	_____
9	Y	_____	_____	_____	_____

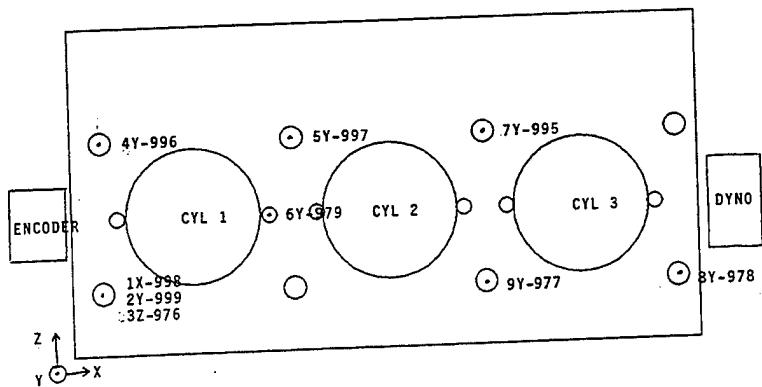
Engine Sensors

Head	Y	_____	_____	_____
Block	Y	_____	_____	_____
Base	Y	_____	_____	_____

Pressure Transducers: (Normally will be cable and channel 1)
 (Normally will only monitor one cylinder)

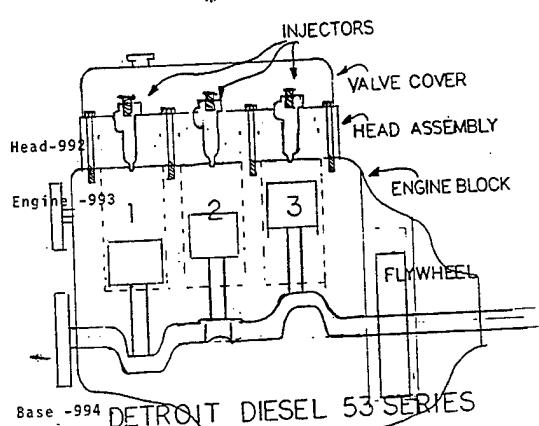
Cylinder	Data Cable (1-4)	SF-891 Channel ECA Interface
1	_____	_____
2	_____	_____
3	_____	_____

Optical Encoder: Ensure connections are tight.



Vibration Sensors:

<u>Serial Number</u>	<u>Scale Factor</u>
976	97.3
977	100.3
978	96.0
979	96.3
992	95.2
993	95.0
994	96.9
995	95.7
996	99.3
997	94.7
998	95.5
999	103.2



APPENDIX C

ENGINE CYCLE ANALYZER SETUP FILES

When the ECA computer boots up the Engine Cycle Analyzer software automatically starts and displays a menu with several choices. The first choice, System Setup will be addressed in this appendix. The other choices, Data Taking, Engine Cycle Analysis, etc... take data or are based on data files previously collected. The basis for these data files is included in the Setup Files. The operation of the other functions are somewhat self explanatory and will not be addressed. Reviewing reference two will acquaint the reader with what ever information is necessary for data processing.

The System Setup selection first prompts the user to select a file. There are currently four files established for data taking in the current system arrangement. The user may select one of them or they may alter one of them and save the changes under that filename or a new name. The setup files are a combination between engine parameters and the sensors to be used to gather data. The common part of the four files is the engine data.

Engine data is entered by the user when establishing a setup file when Engine Design is selected. The data requested is explained in reference 2 page 3-26, but specifically is the following:

Engine Model	In line Diesel 3 Cylinder
Bore	B=3.875
Stroke	S=4.5
ConRod	L=8.8
Offset	d=0.0
Comp Ratio	CR=21
Inl Close	IC=90
Exh Open	EO=85

Spk/Inj	SPK=N
Vor I type	<V,I>=I
No of Cylinders	n=3

These will be the same for all configurations unless the engine is changed.

The other menu sections which make changes to the system are INSTRUMENT 1, INSTRUMENT 2, and OPER CONDITION. The INSTRUMENT 1 menu selection is information on the type and parameters of the optical encoder. For all applications, unless the optical encoder is changed, the following information will hold true.

A. Number of Strokes	2
B. Trigger, External	E
1. External	
Encoder Resolution	0.1
Sample Resolution	0.1
2. Internal	
Encoder Resolution	0.1

File 720DD535.SET is written for the 720 count optical encoder. All other file assume the 3600 count encoder is installed.

The menu selection INSTRUMENT 2 changes the sensors which will be sampled by the ECA system. It defines which channels will take which data, and what scaling factors are used to convert the required 5 volt or less signal to a usable unit or quantity. The input screen allows the user to define the four channels of data acquisition. The user may input the channel ID to describe the sensor. The signal unit depends on the sensor, pressure sensors measure in bars, which must be converted to psi in the scale factor b, vibration sensor use units of g's. All of the current sensors are linear and have no Voltage offset a, which should be set to zero. The scale factors which may be input are the following for the appropriate sensor. The vibration sensors are listed by serial number as their locations may change.

Vibration Sensors

<u>Serial Number</u>	<u>Scale Factor</u>
976	97.3
977	100.3
978	96.0
979	96.3
992	95.2
993	95.0
994	96.9
995	95.7
996	99.3
997	94.7
998	95.5
999	103.2

Pressure Transducers

<u>Serial Number</u>	<u>Scale Factor</u>
570205	11436
570206	11612
570162	11600

The following files have the following sensors preset:

CH	DD353.SET	DD353PRE.SET	53ENGVIB.SET	353VIBES.SET
1	Press Cyl 1	Press Cyl 1	Vibes Base	Vibes 7Y
2	Vibes 2Y	Press Cyl 2	Vibes Eng	Vibes 2Y
3	Vibes 8Y	Press Cyl 3	Vibes Head	Vibes 8Y
4	Vibes 5Y	Vibes 5Y	Vibes 2Y	Vibes 5Y

The last part of the setup process is the OPERATION CONDITIONS. The quantities A through G are self explanatory and do not affect the calculations of the ECA and are reference purposes only. Input H is for Engine Load BMEP (Brake Mean Effective Pressure). The maximum rated BMEP of the engine as installed is 83psi. The BMEP changes with the brake torque acting on the engine. The following equation is used to compute BMEP:

$$BMEP = \frac{CN_p T_B}{disp}$$

G-1

Where C is a conversion constant equal to 75.4. N_p is the number of revolutions per power stroke which is one for a two stroke engine. T_B is the brake torque for the test measured in ft-lbs. The displacement of the engine is represented by disp and equals 159in³. [Ref. 15]

Engine Design & Test Input File Summary

File Name: DD353.SET Print Date: 8/18/1996
Engine ID: I3-53.2D Test ID:
Unit System Used: Customary

Engine Design

Engine Model: Inline Diesel 3 Cyl
B X S = 3.880 X 4.500 L = 8.800
CR = 21.0 R/L = 0.26
CID/n = 53.2
IC = 90.0 EO = 85.0
d = 0.00 d/B = 0.00

Operating Condition

BMEP = 0 RPM = Set F/A = 1.000
Amb Pres = Inl MAP = 1.0 Ext MAP = 1.0
Amb Temp = Inl MAT = 100.0 Ext MAT =
Spk or Inj Timing =

Signal Information

Encoder Resolution = 0.1 Sample Resolution = 0.1

	Signal	Unit	Scale	Offset
CH 1	Pcyl1	psi	11612.0	0.00
CH 2	Vib2y	g	103.2	0.00
CH 3	Vib8y	g	96.0	0.00
CH 4	Vib5y	g	94.7	0.00

Engine Design & Test Input File Summary

File Name: 53ENGVIB.SET Print Date: 8/18/1996
Engine ID: I3-53.2D Test ID:
Unit System Used: Customary

Engine Design

Engine Model: Inline Diesel 3 Cyl
B X S = 3.880 X 4.500 L = 8.800
CR = 21.0 R/L = 0.26
CID/n = 53.2
IC = 90.0 EO = 85.0
d = 0.00 d/B = 0.00

Operating Condition

BMEP = 0 RPM = Set F/A = 1.000
Amb Pres = Inl MAP = 1.0 Ext MAP = 1.0
Amb Temp = Inl MAT = 100.0 Ext MAT =
Spk or Inj Timing =

Signal Information

Encoder Resolution = 0.1 Sample Resolution = 0.1

	Signal	Unit	Scale	Offset
CH 1	Base	g	96.9	0.00
CH 2	Engine	g	95.1	0.00
CH 3	Head	g	95.2	0.00
CH 4	Vib2y	g	103.2	0.00

Engine Design & Test Input File Summary

File Name: DD353PRE.SET Print Date: 8/18/1996
Engine ID: I3-53.2D Test ID:
Unit System Used: Customary

Engine Design

Engine Model: Inline Diesel 3 Cyl
B X S = 3.880 X 4.500 L = 8.800
CR = 21.0 R/L = 0.26
CID/n = 53.2
IC = 90.0 EO = 85.0
d = 0.00 d/B = 0.00

Operating Condition

BMEP = 0 RPM = Set F/A = 1.000
Amb Pres = Inl MAP = 1.0 Ext MAP = 1.0
Amb Temp = Inl MAT = 100.0 Ext MAT =
Spk or Inj Timing =

Signal Information

Encoder Resolution = 0.1 Sample Resolution = 0.1

	Signal	Unit	Scale	Offset
CH 1	Pcyl1	psi	11612.0	0.00
CH 2	pcyl2	psi	11436.0	0.00
CH 3	pcyl3	psi	11600.0	0.00
CH 4	Vib5y	g	94.7	0.00

Engine Design & Test Input File Summary

File Name: 53VIBES.SET Print Date: 8/18/1996
Engine ID: I3-53.2D Test ID:
Unit System Used: Customary

Engine Design

Engine Model: Inline Diesel 3 Cyl
B X S = 3.880 X 4.500 L = 8.800
CR = 21.0 R/L = 0.26
CID/n = 53.2
IC = 90.0 EO = 85.0
d = 0.00 d/B = 0.00

Operating Condition

BMEP = 0 RPM = Set F/A = 1.000
Amb Pres = Inl MAP = 1.0 Ext MAP = 1.0
Amb Temp = Inl MAT = 100.0 Ext MAT =
Spk or Inj Timing =

Signal Information

Encoder Resolution = 0.1 Sample Resolution = 0.1

	Signal	Unit	Scale	Offset
CH 1	Vib7y	g	95.7	0.00
CH 2	Vib2y	g	103.2	0.00
CH 3	Vib8y	g	96.0	0.00
CH 4	Vib5y	g	94.7	0.00

APPENDIX D

JTFA VIBRATION SIGNALS

The following figures are vibration and JTFA signals from vibration transducers installed on the engine. This data is meant to be used for historical reference. It may be considered the baseline data for the engine. It is sorted by head bolt torque, data run, and engine conditions.

D.1 HEAD BOLT TORQUE 180 ft-lb

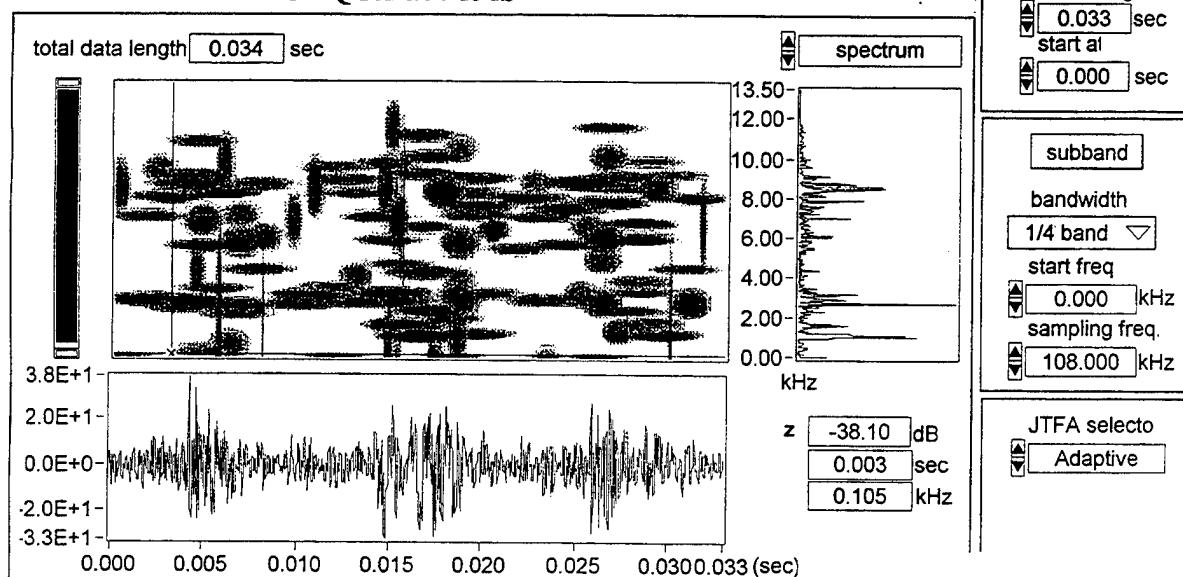


Figure D.1 Engine Transducer 1800 RPM 125 ft-lb

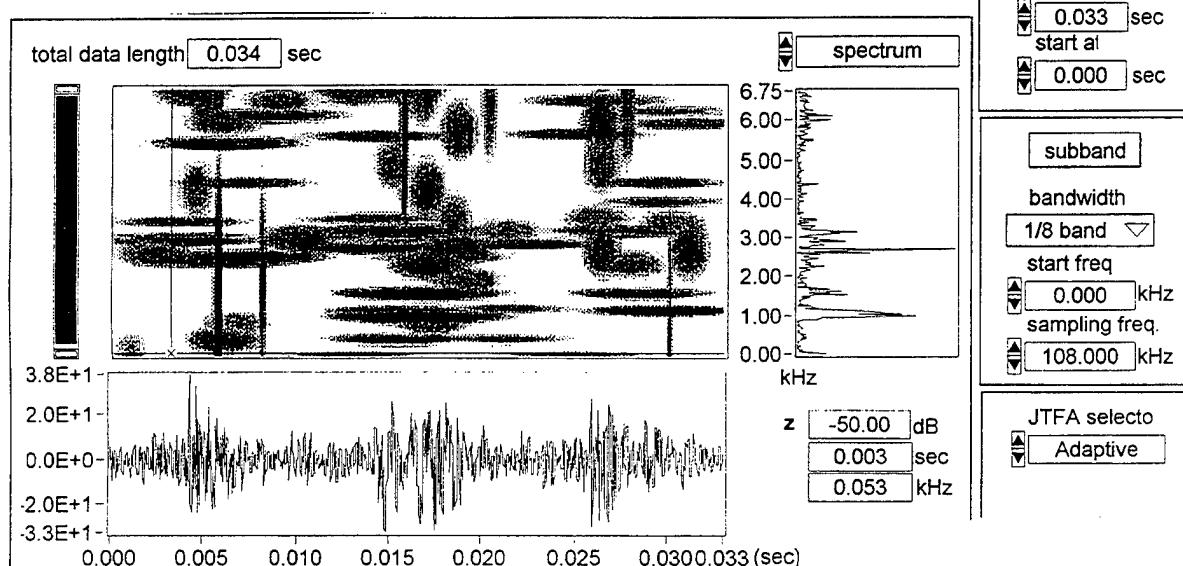


Figure D.2 Engine Transducer 1800 RPM 125 ft-lb

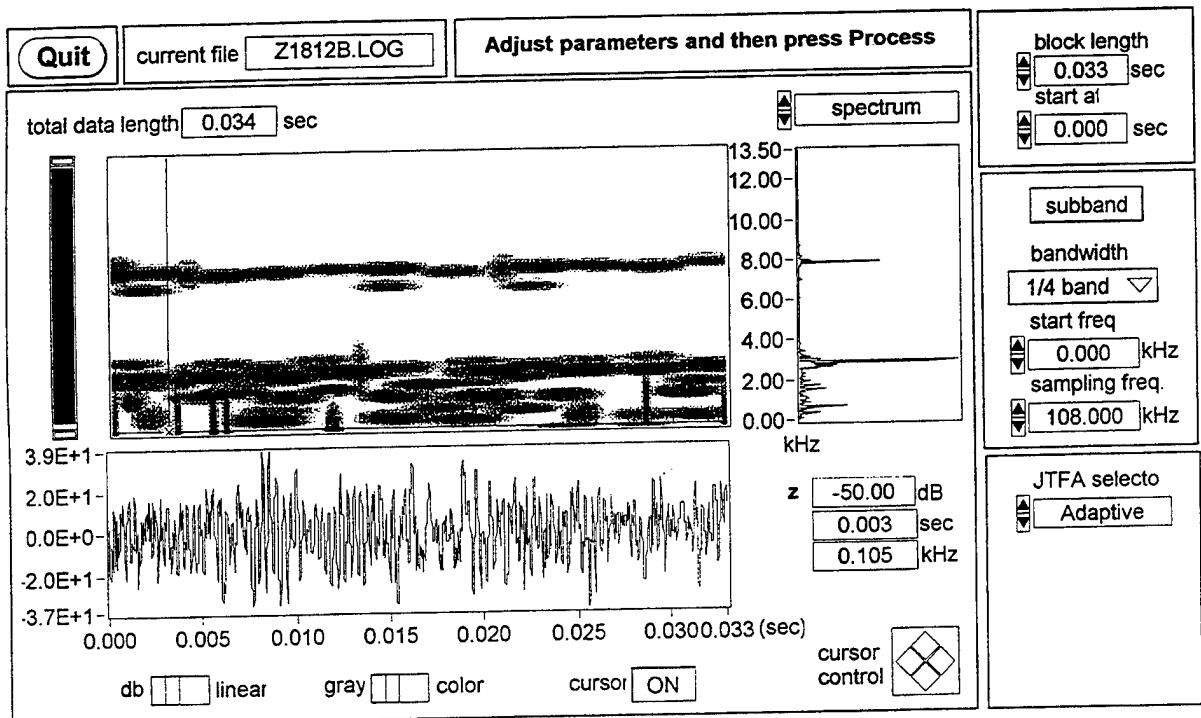


Figure D.3 Base Transducer 1800 RPM 125 ft-lb

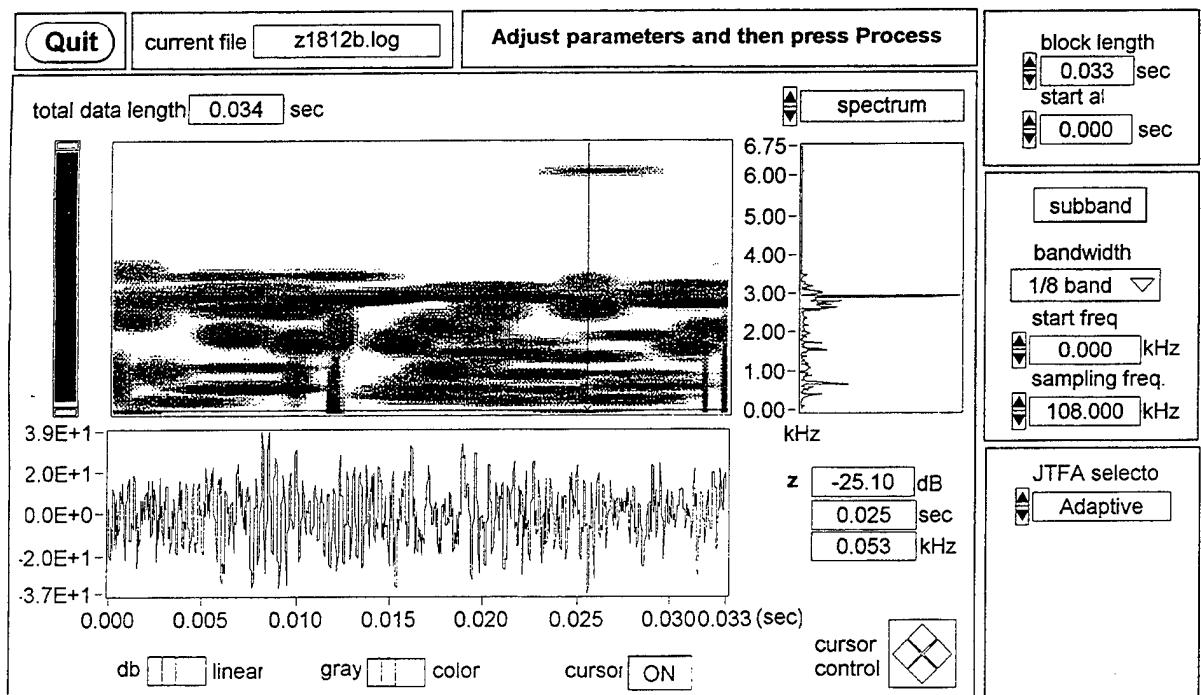


Figure D.4 Base Transducer 1800 RPM 125 ft-lb

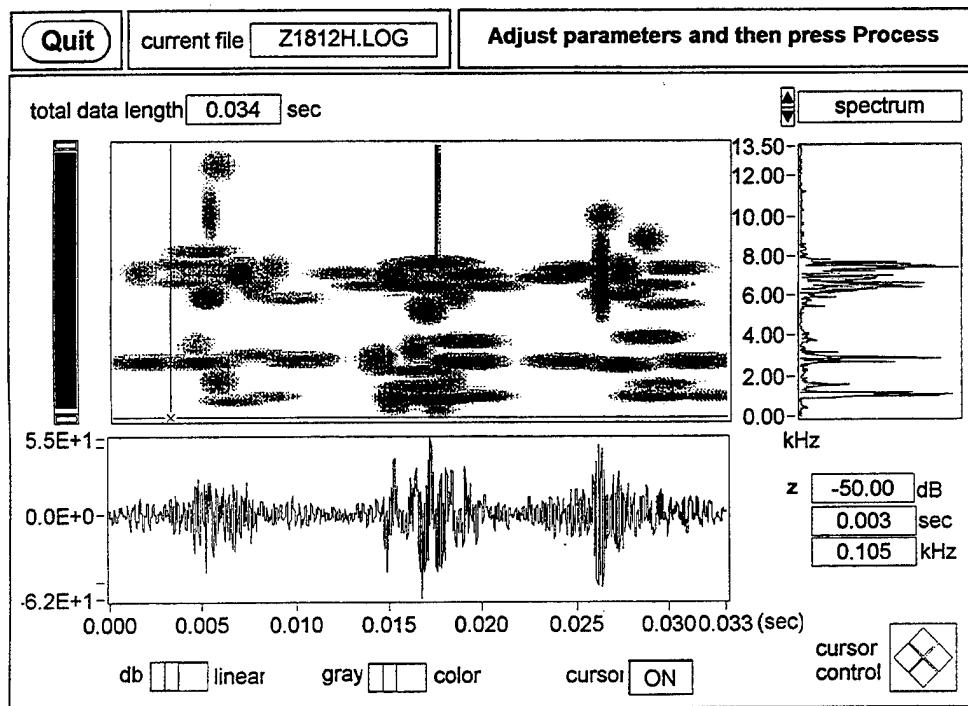


Figure D.5 Head Assembly Transducer 1800 RPM 125 ft-lb

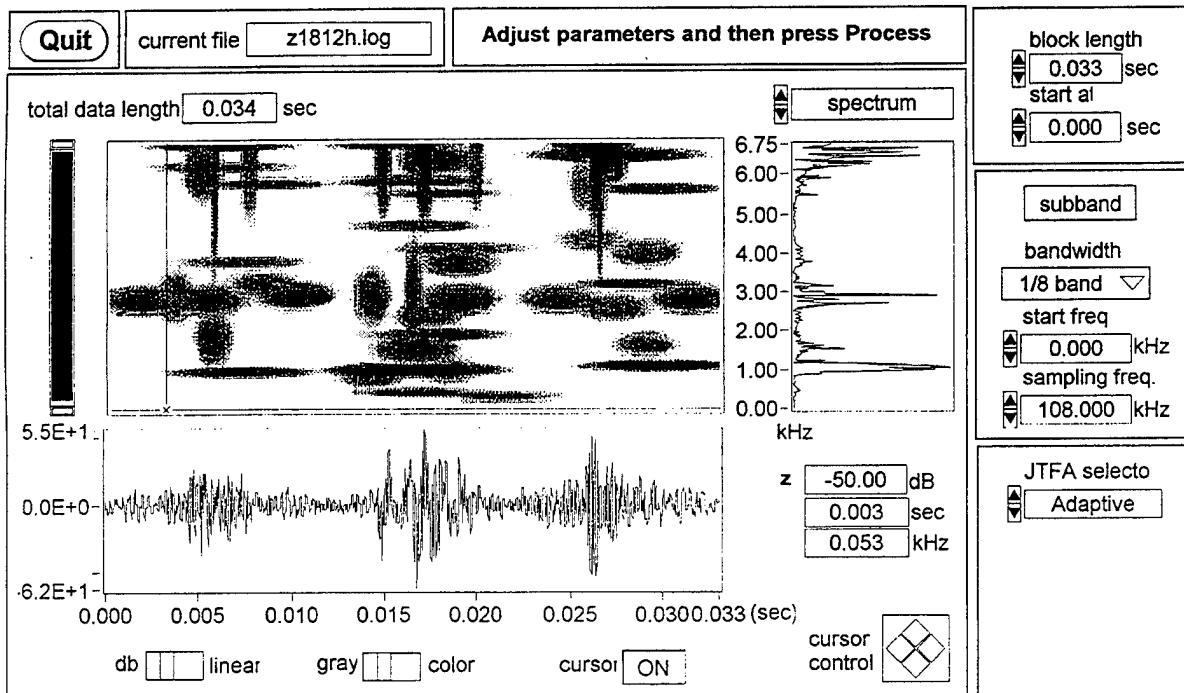


Figure D.6 Head Assembly Transducer 1800 RPM 125 ft-lb

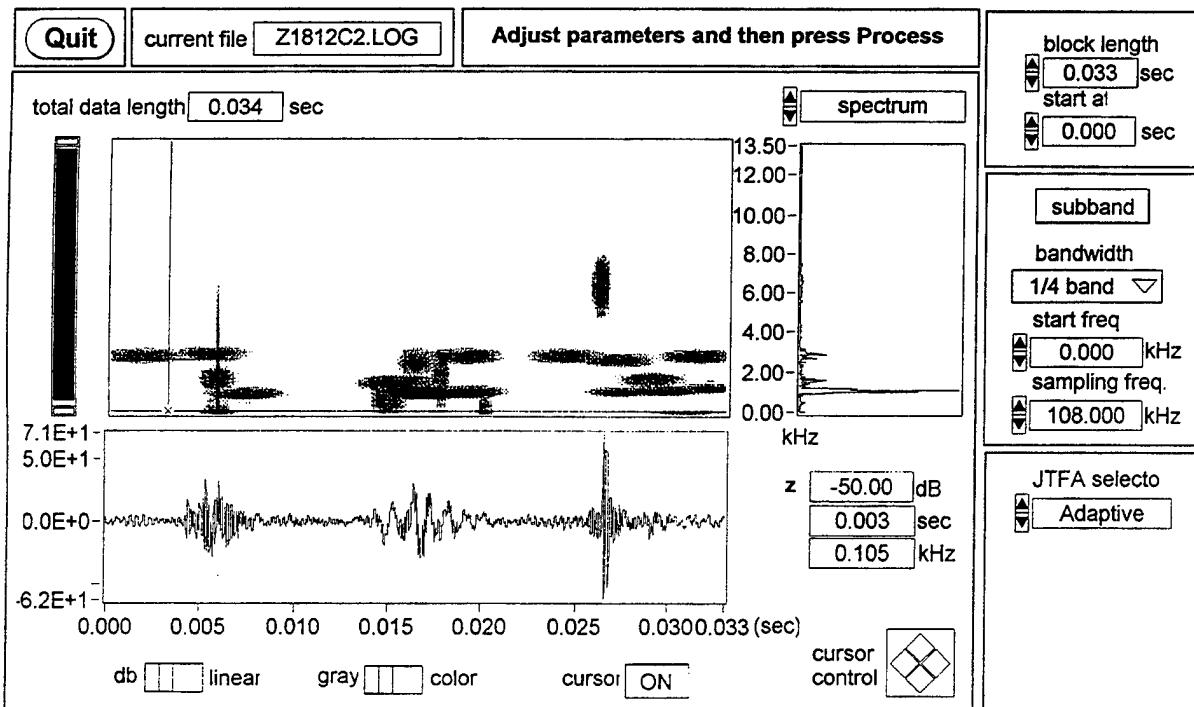


Figure D.7 Head Bolt Transducer C2 1800 RPM 125 ft-lb

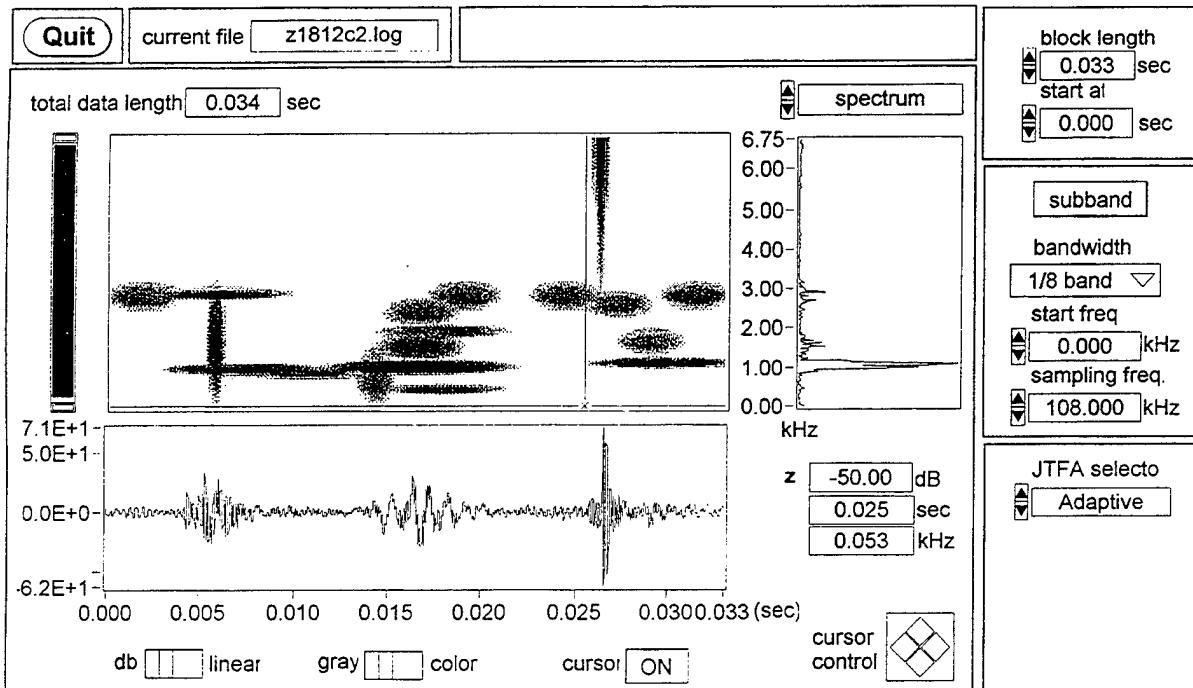


Figure D.8 Head Bolt Transducer C2 1800 RPM 125 ft-lb

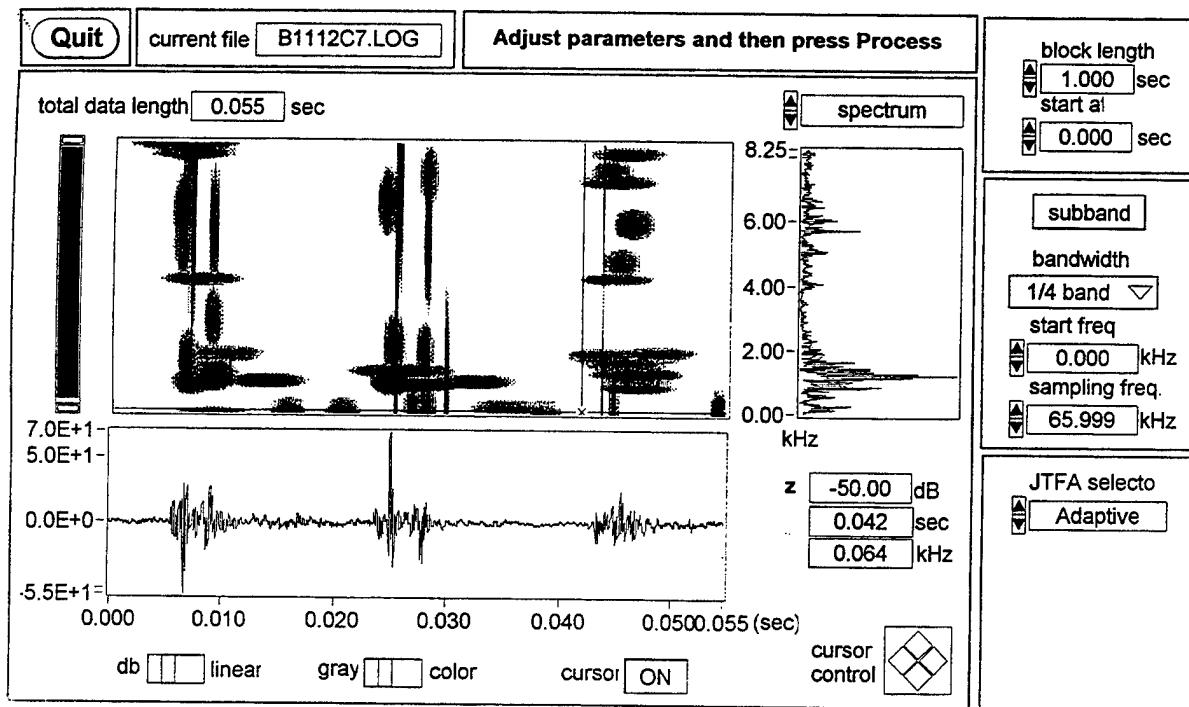


Figure D.9 Head Bolt Transducer C7 1100 RPM 125 ft-lb

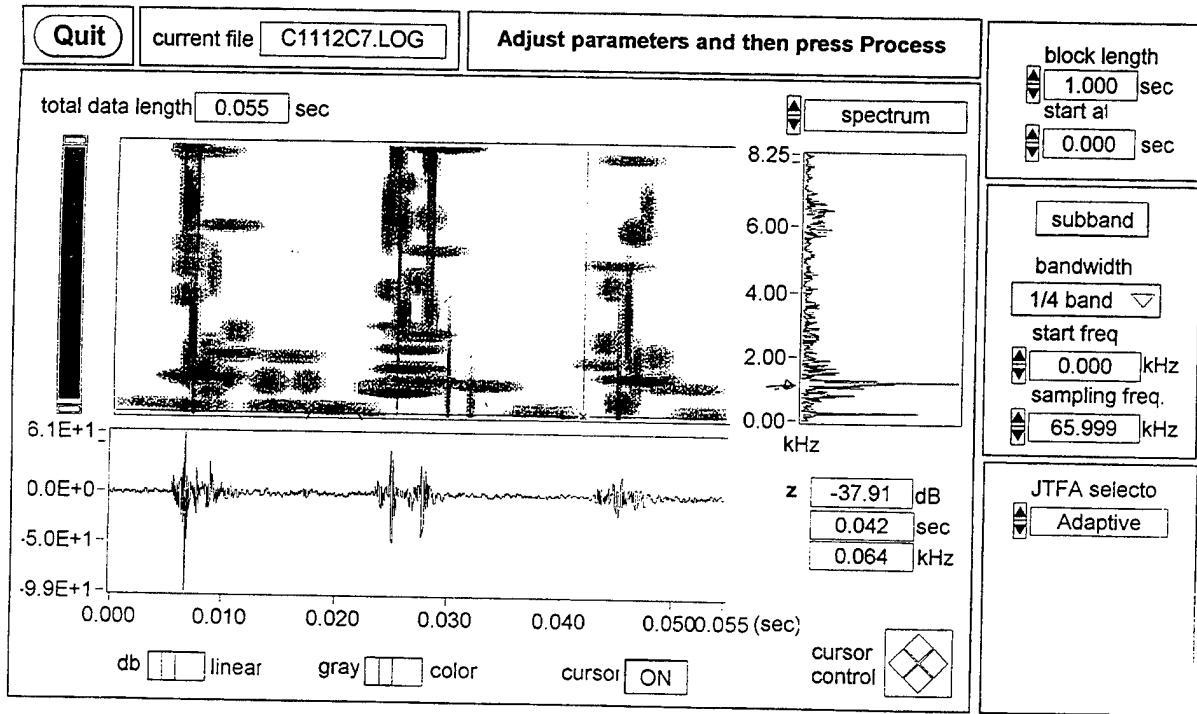


Figure D.10 Head Bolt Transducer C7 1100 RPM 125 ft-lb

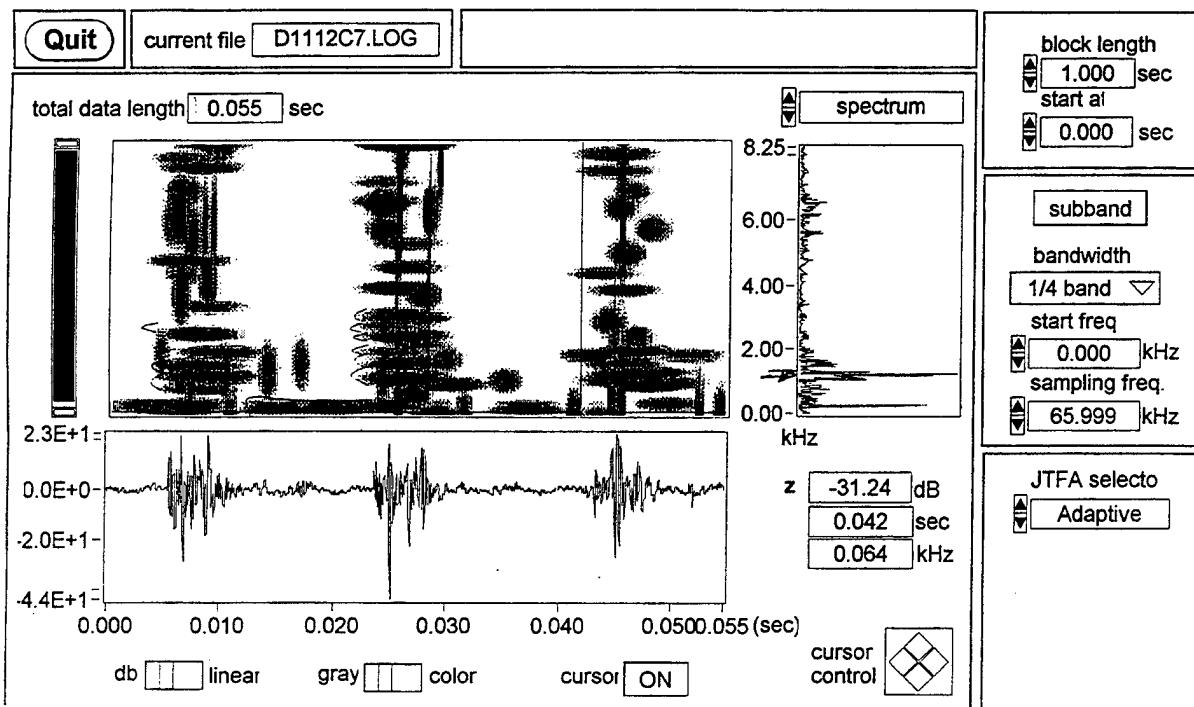


Figure D.11 Head Bolt Transducer C7 1100 RPM 125 ft-lb

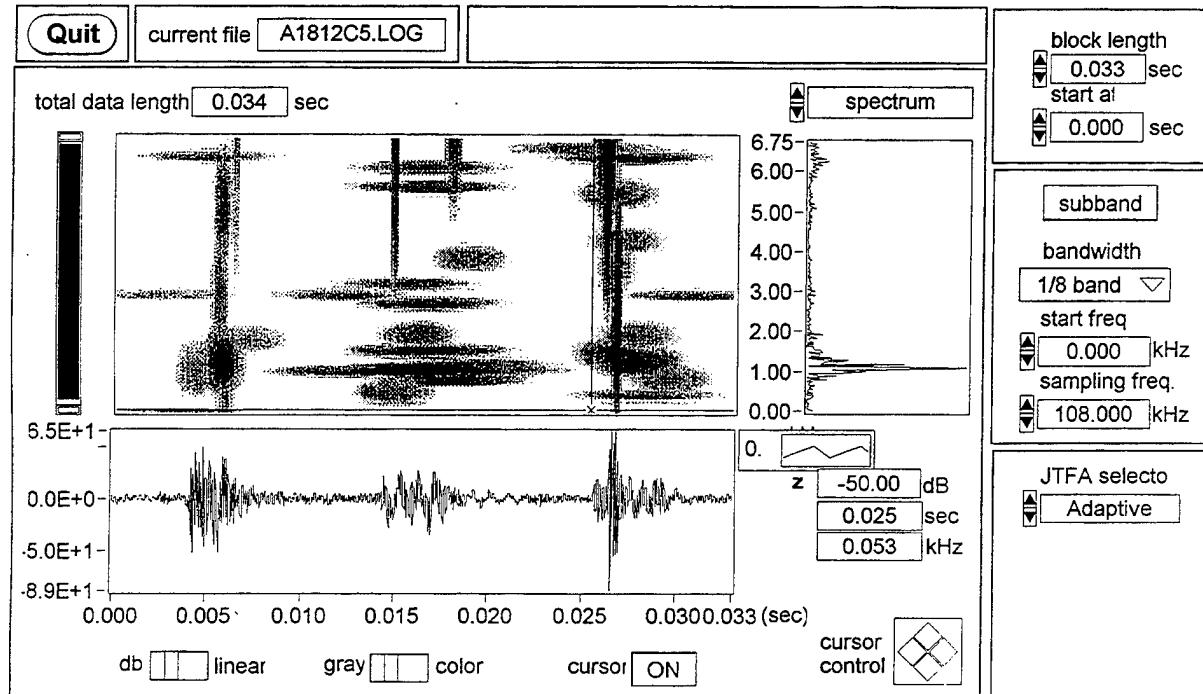


Figure D.12 Head Bolt Transducer C5 1800 RPM 125 ft-lb

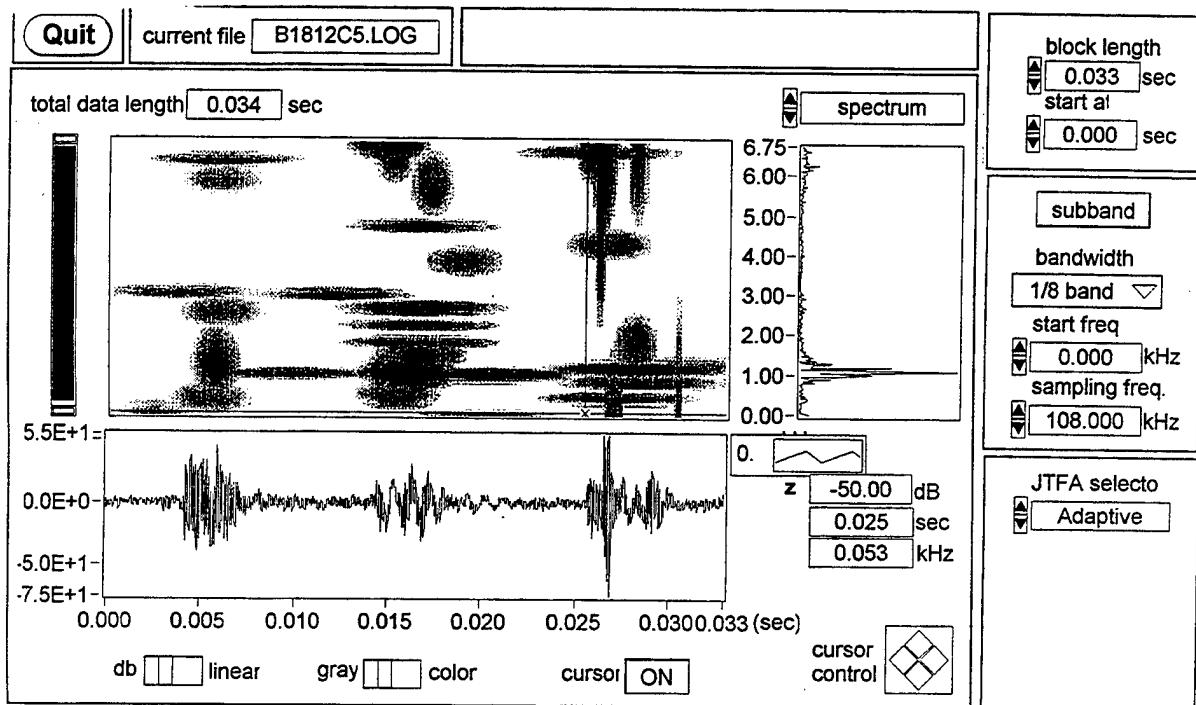


Figure D.13 Head Bolt Transducer C5 1800 RPM 125 ft-lb

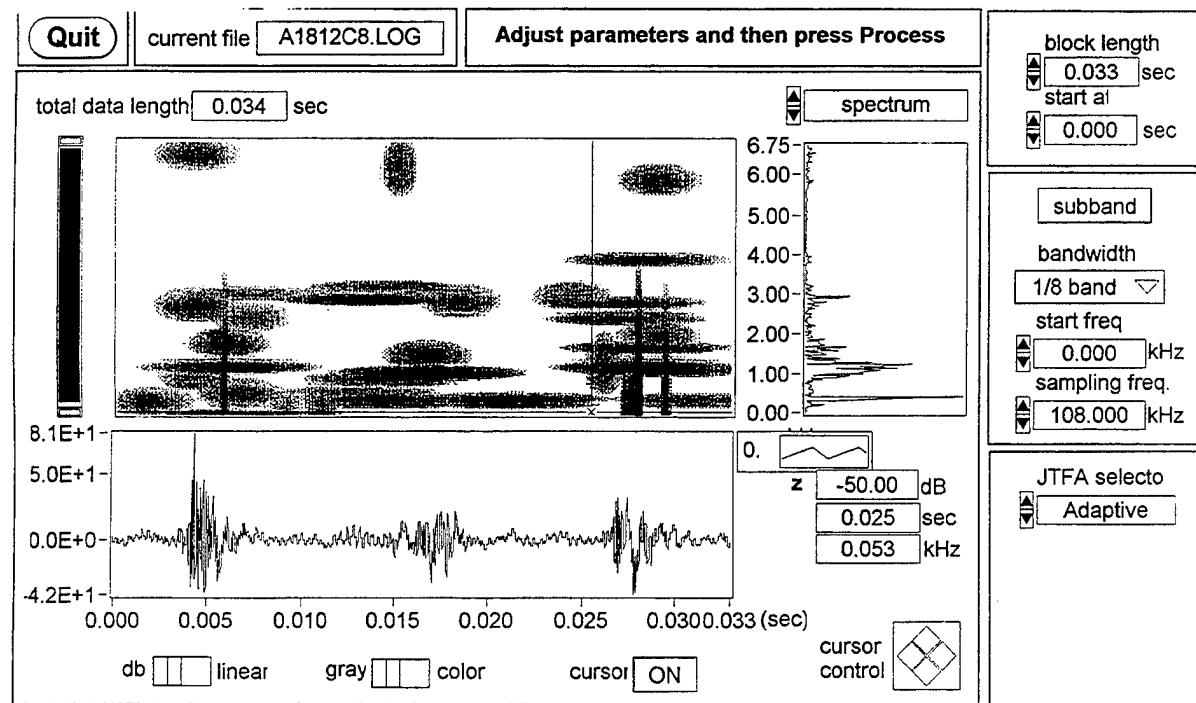


Figure D.14 Head Bolt Transducer C8 1800 RPM 125 ft-lb

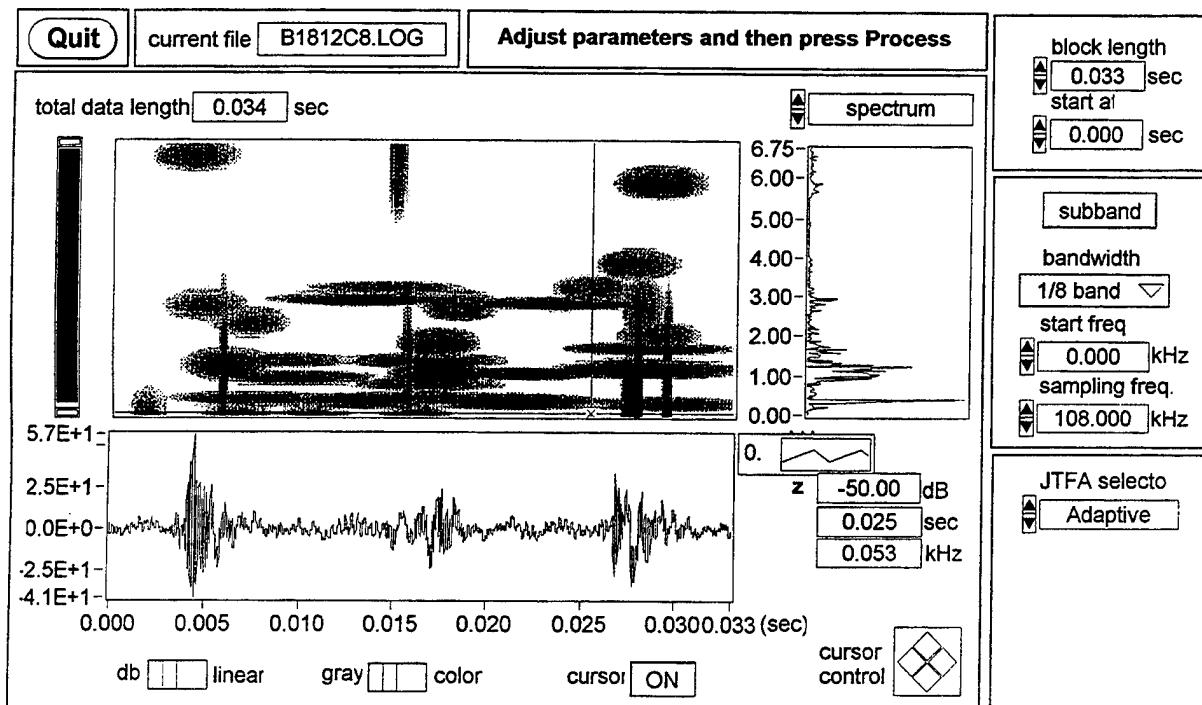


Figure D.15 Head Bolt Transducer C8 1800 RPM 125 ft-lb

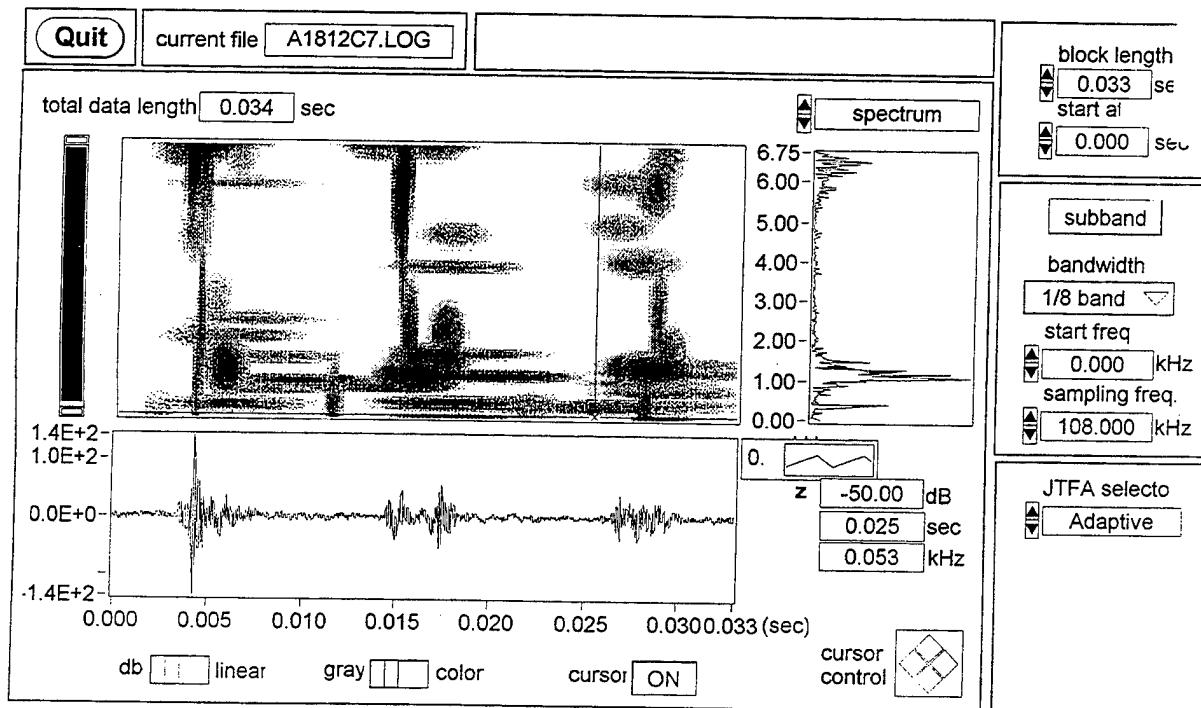


Figure D.16 Head Bolt Transducer C7 1800 RPM 125 ft-lb

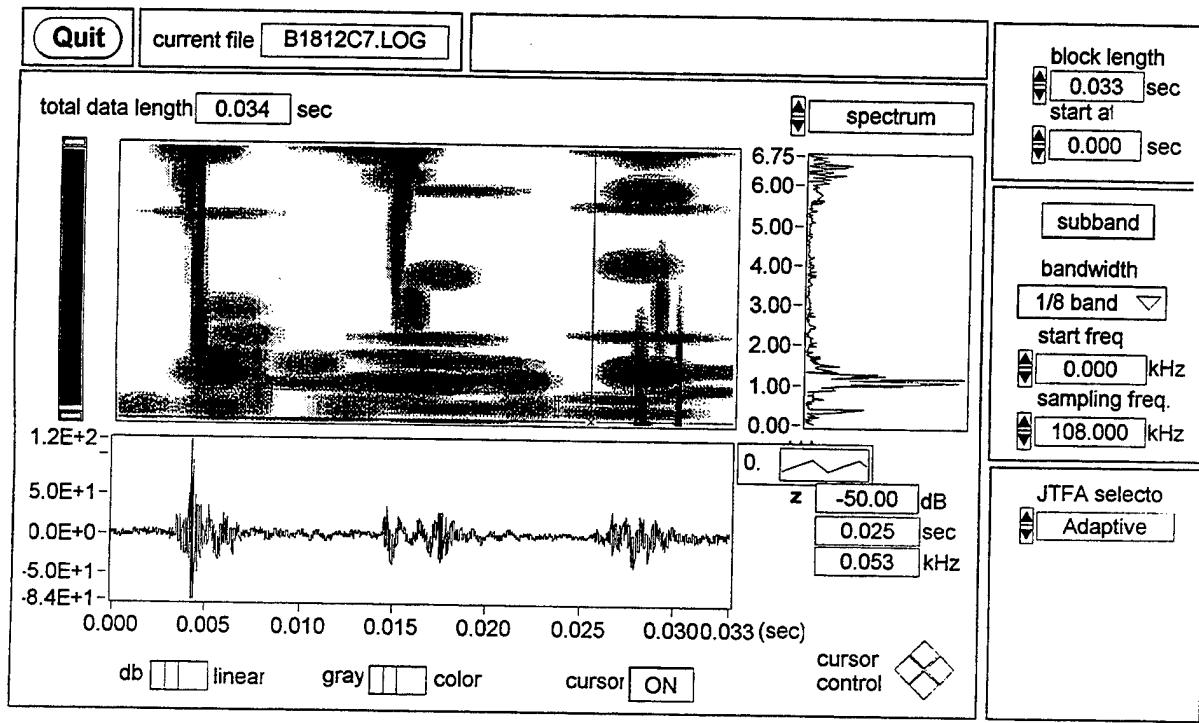


Figure D.17 Head Bolt Transducer C7 1800 RPM 125 ft-lb

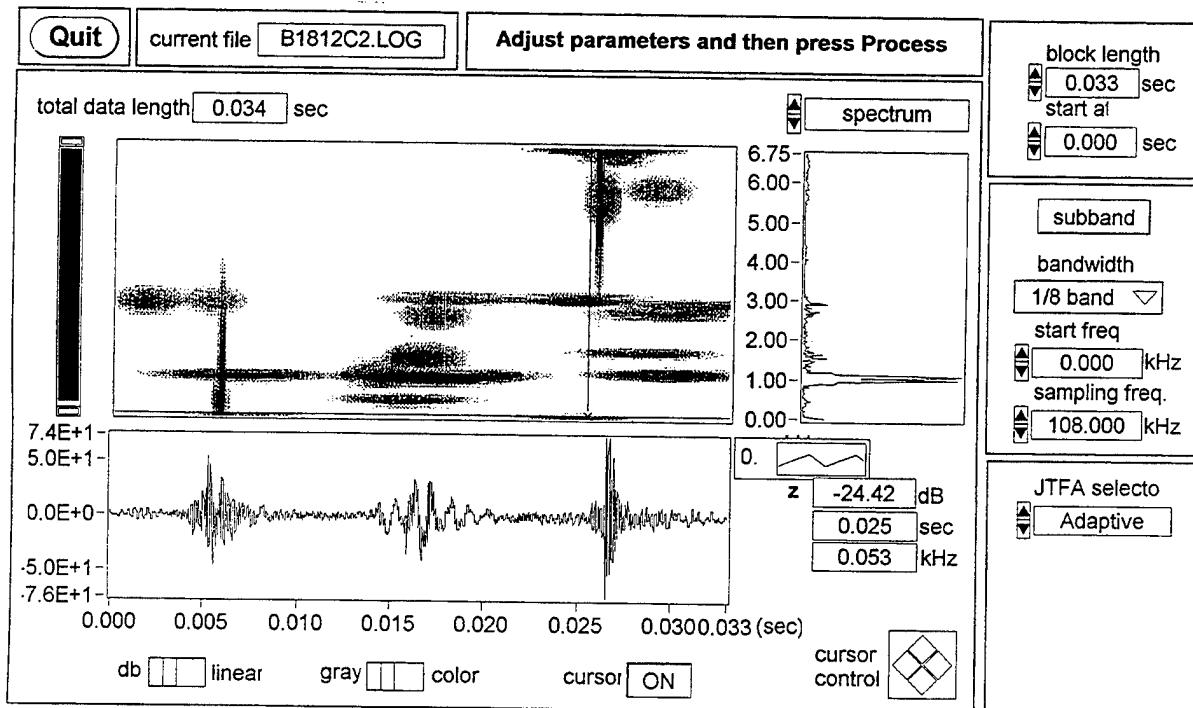


Figure D.18 Head Bolt Transducer C2 1800 RPM 125 ft-lb

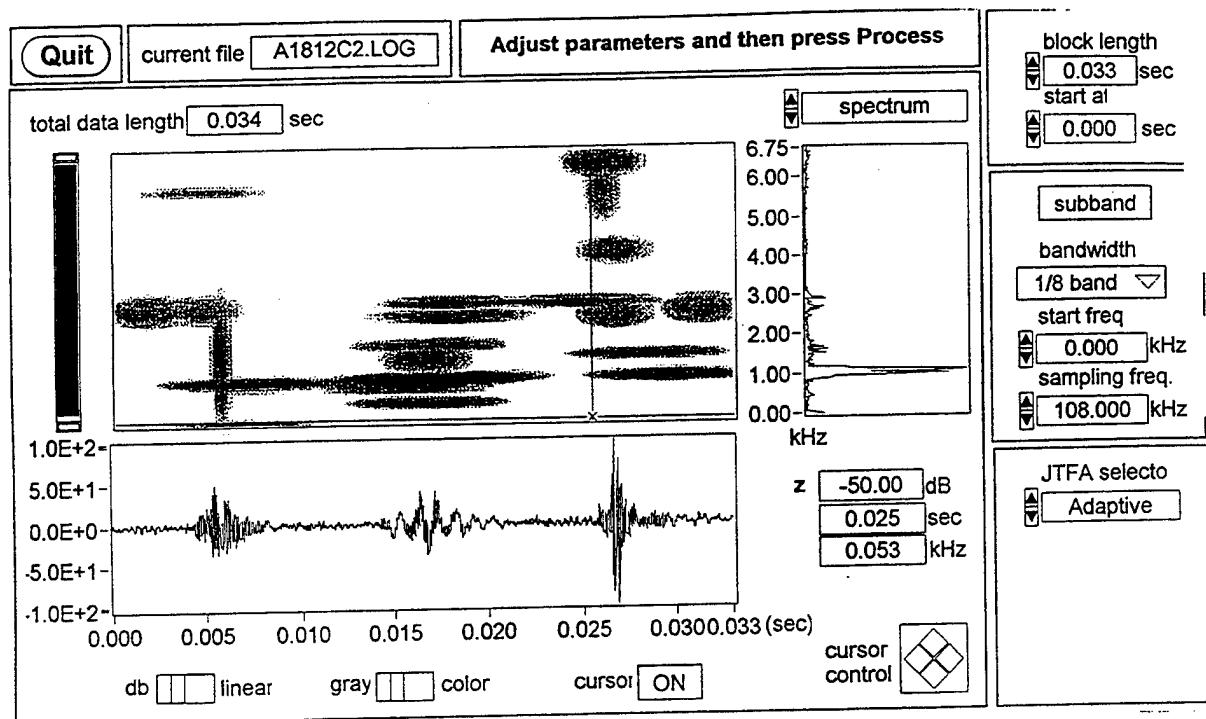


Figure D.19 Head Bolt Transducer C2 1800 RPM 125 ft-lb

D.2 HEAD BOLT TORQUE 160 ft-lb

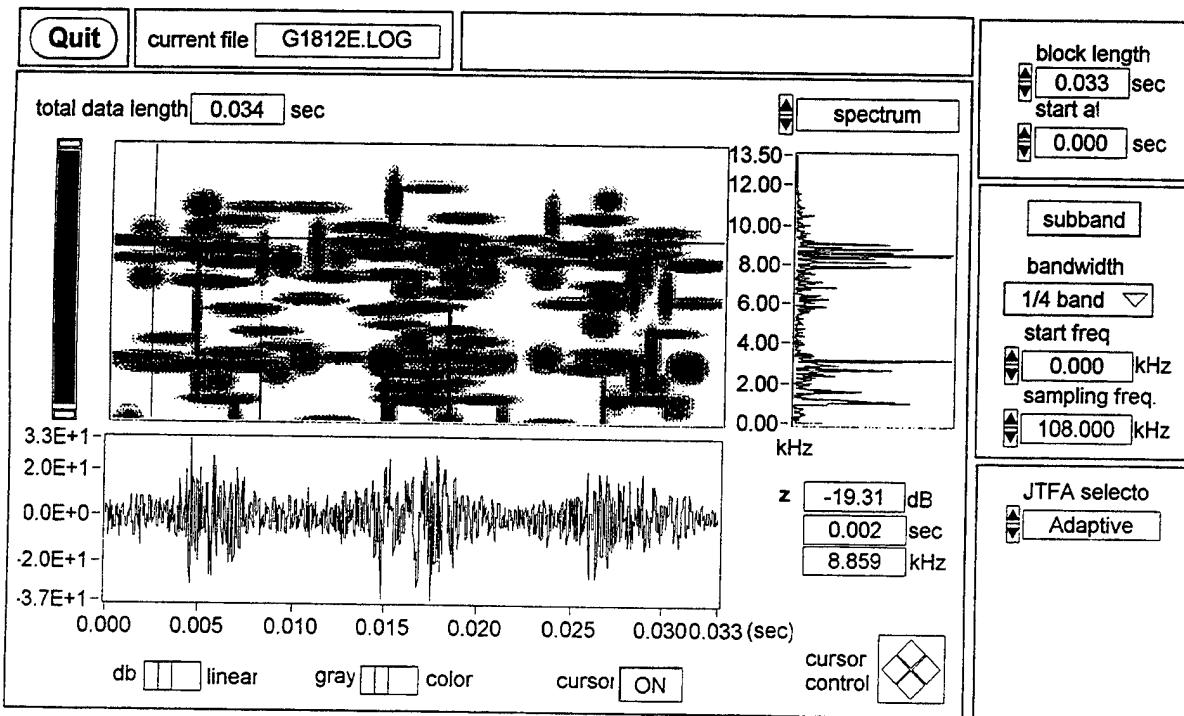


Figure D.20 Engine Transducer 1800 RPM 125 ft-lb

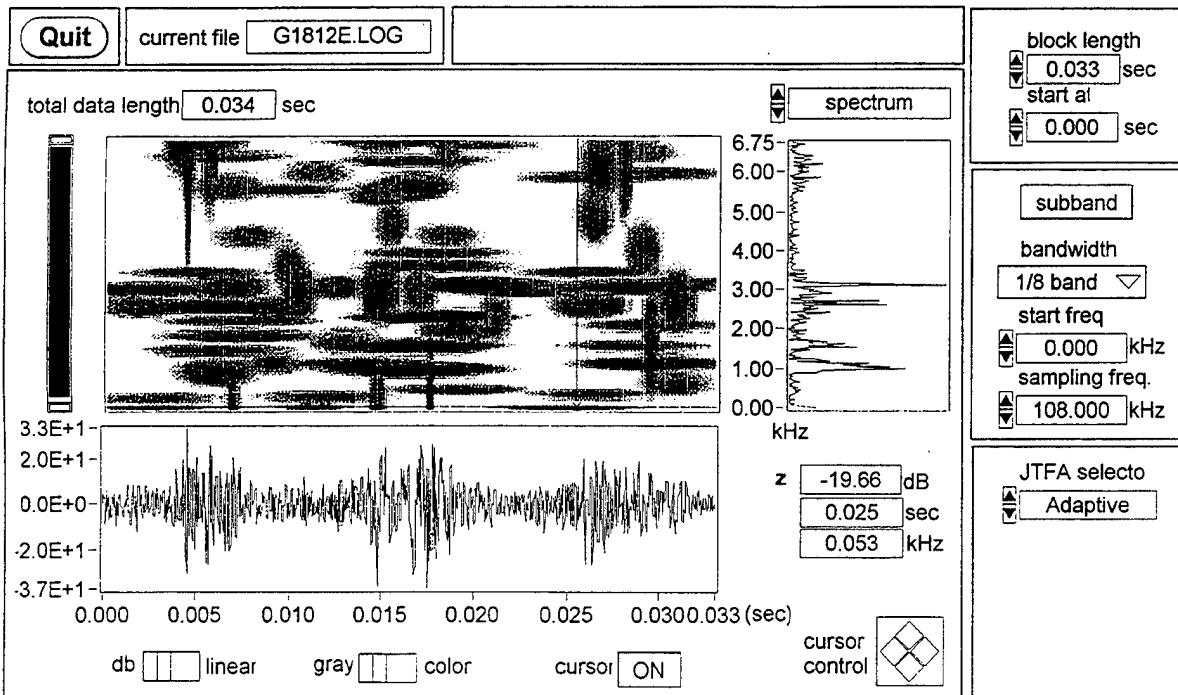


Figure D.21 Engine Transducer 1800 RPM 125 ft-lb

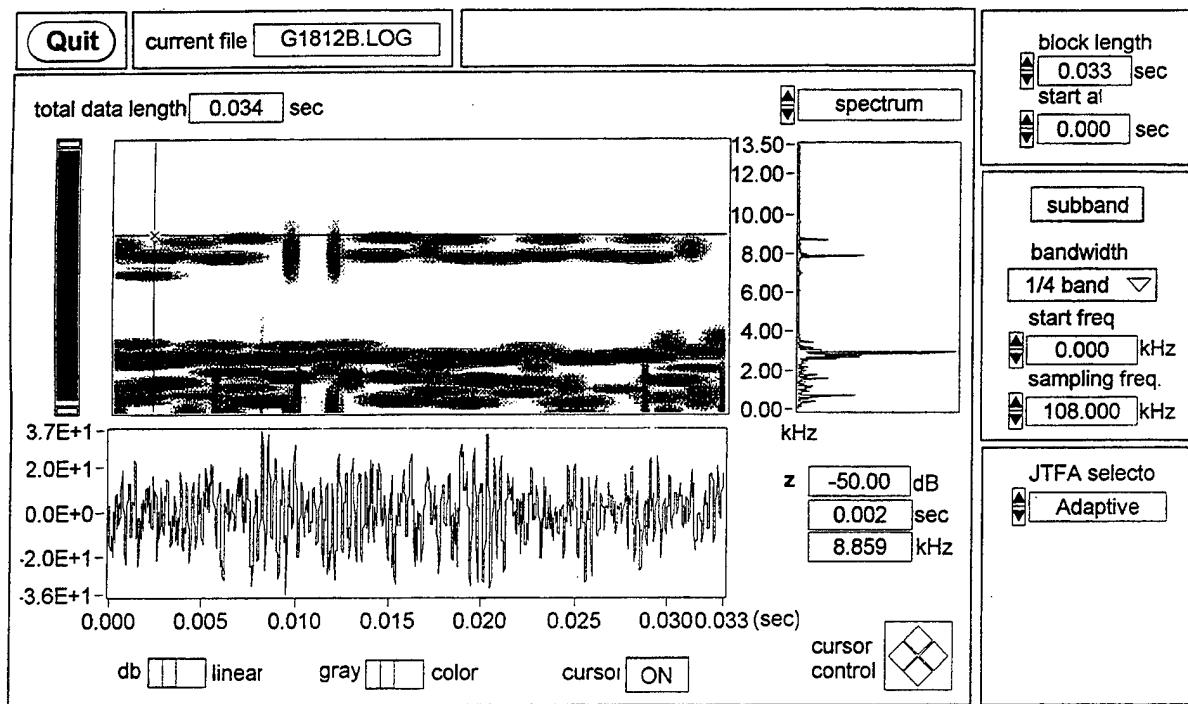


Figure D.22 Engine Base Transducer 1800 RPM 125 ft-lb

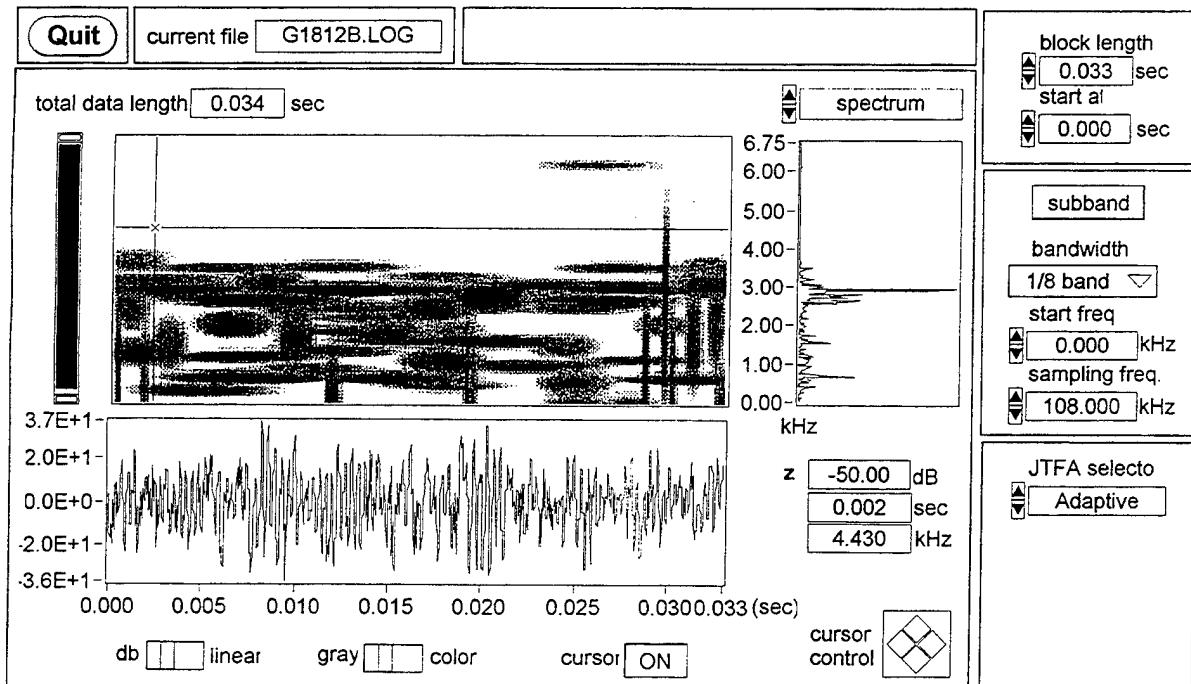


Figure D.23 Engine Base Transducer 1800 RPM 125 ft-lb

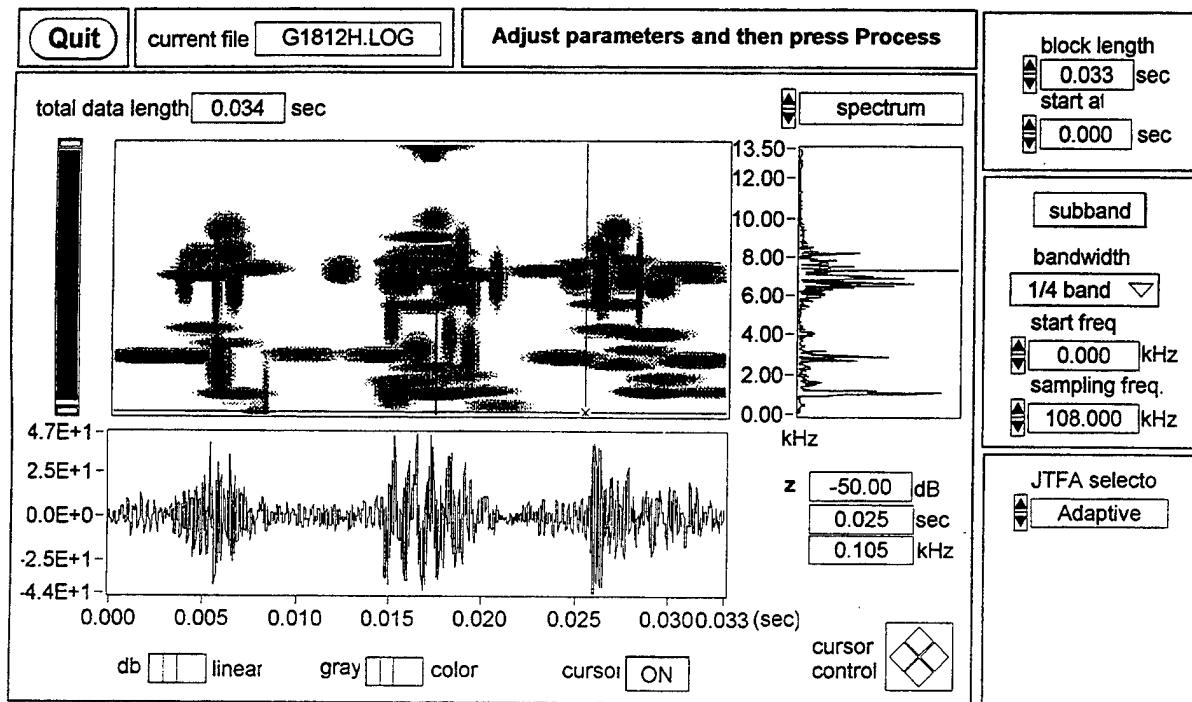


Figure D.24 Head Assembly Transducer 1800 RPM 125 ft-lb

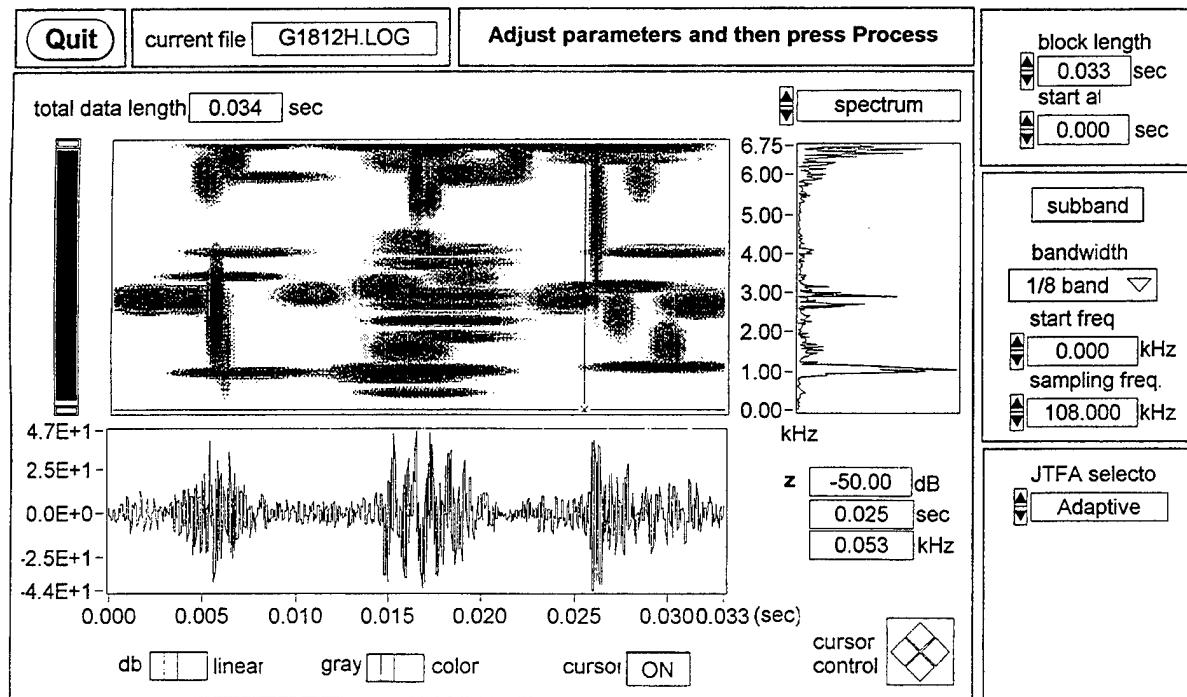


Figure D.25 Head Assembly Transducer 1800 RPM 125 ft-lb

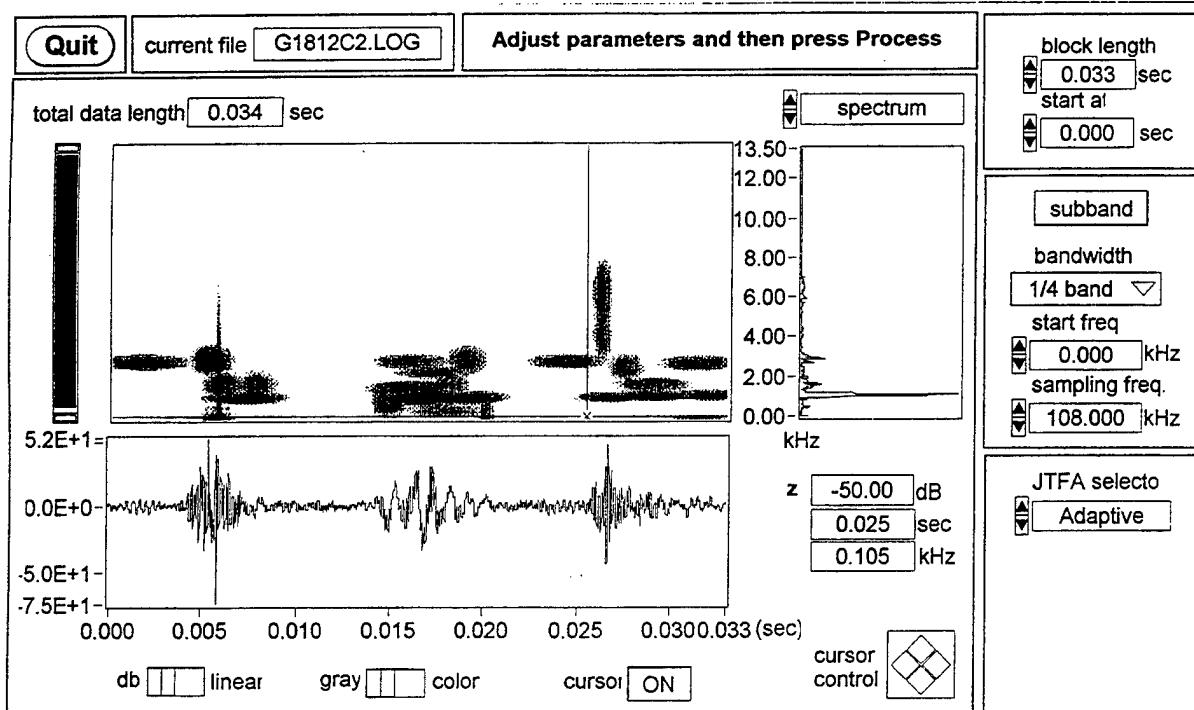


Figure D.26 Head Bolt Transducer C2 1800 RPM 125 ft-lb

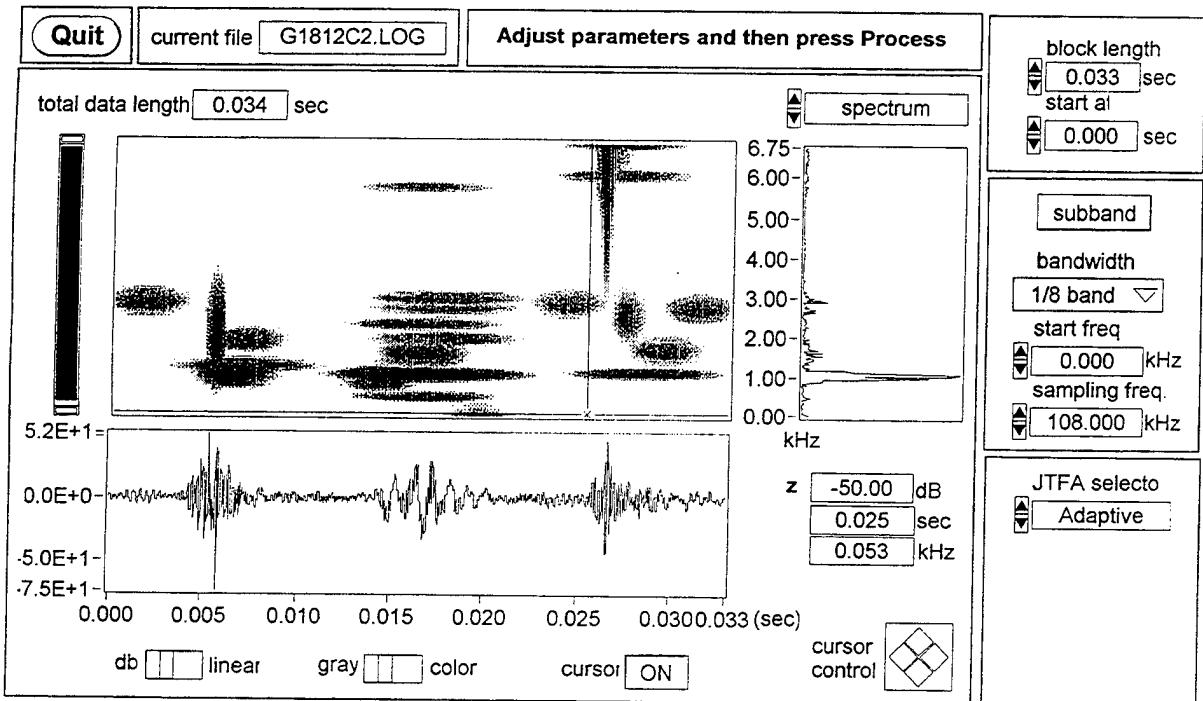


Figure D.27 Head Bolt Transducer C2 1800 RPM 125 ft-lb

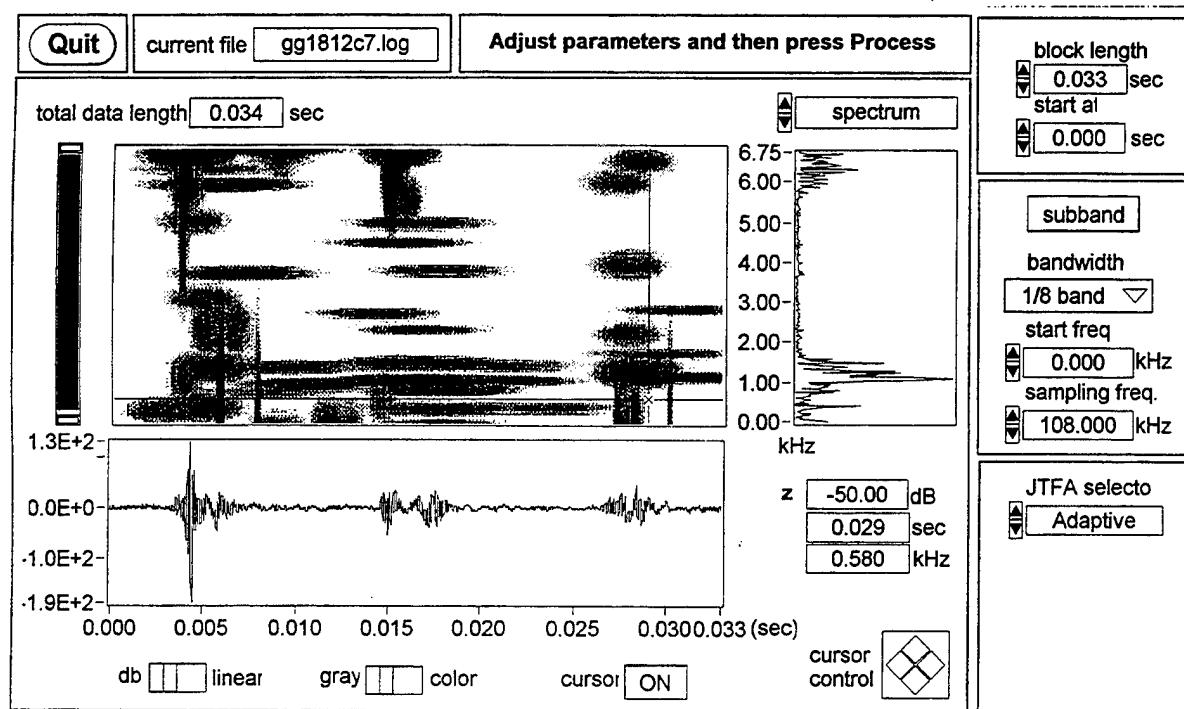


Figure D.28 Head Bolt Transducer C7 1800 RPM 125 ft-lb

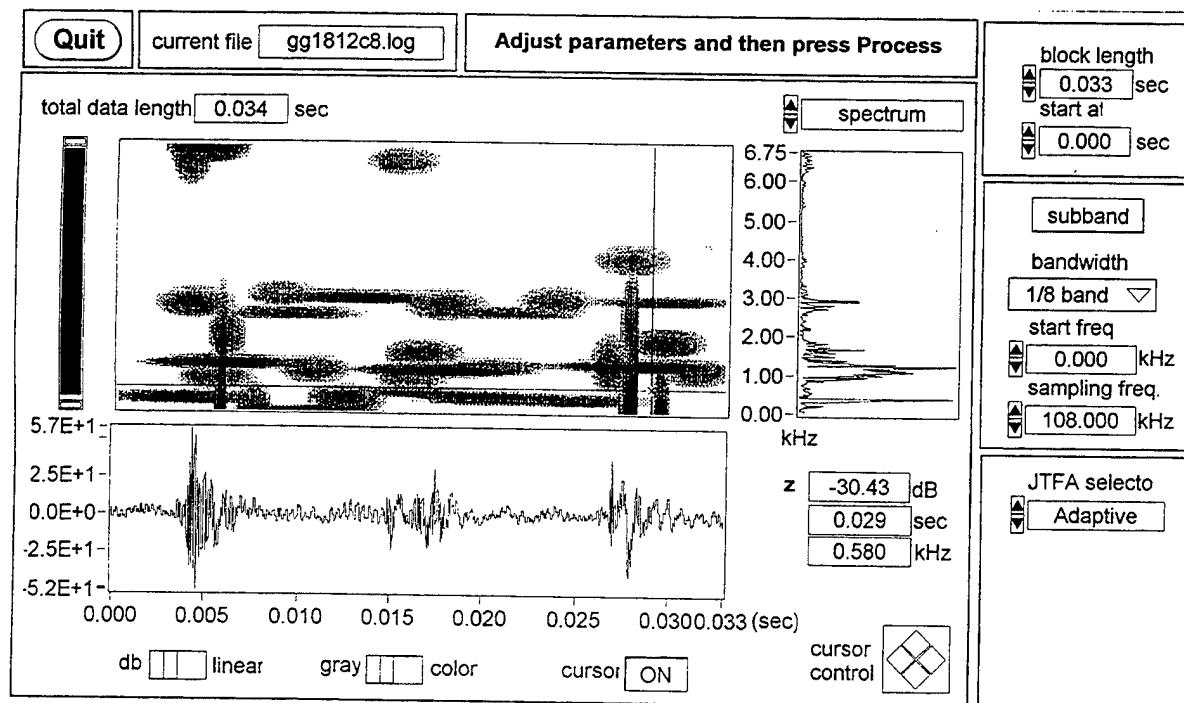


Figure D.29 Head Bolt Transducer C8 1800 RPM 125 ft-lb

D.3 HEAD BOLT TORQUE 190 ft-lb

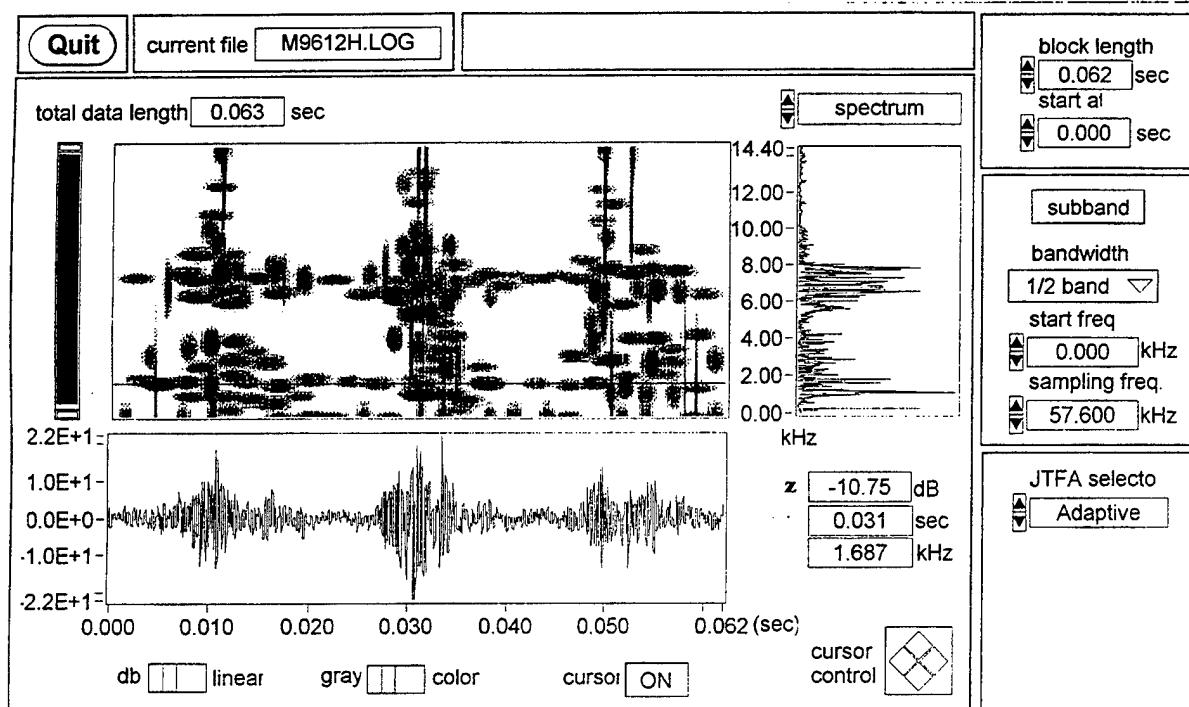


Figure D.30 Head Assembly Transducer 950 RPM 125 ft-lb

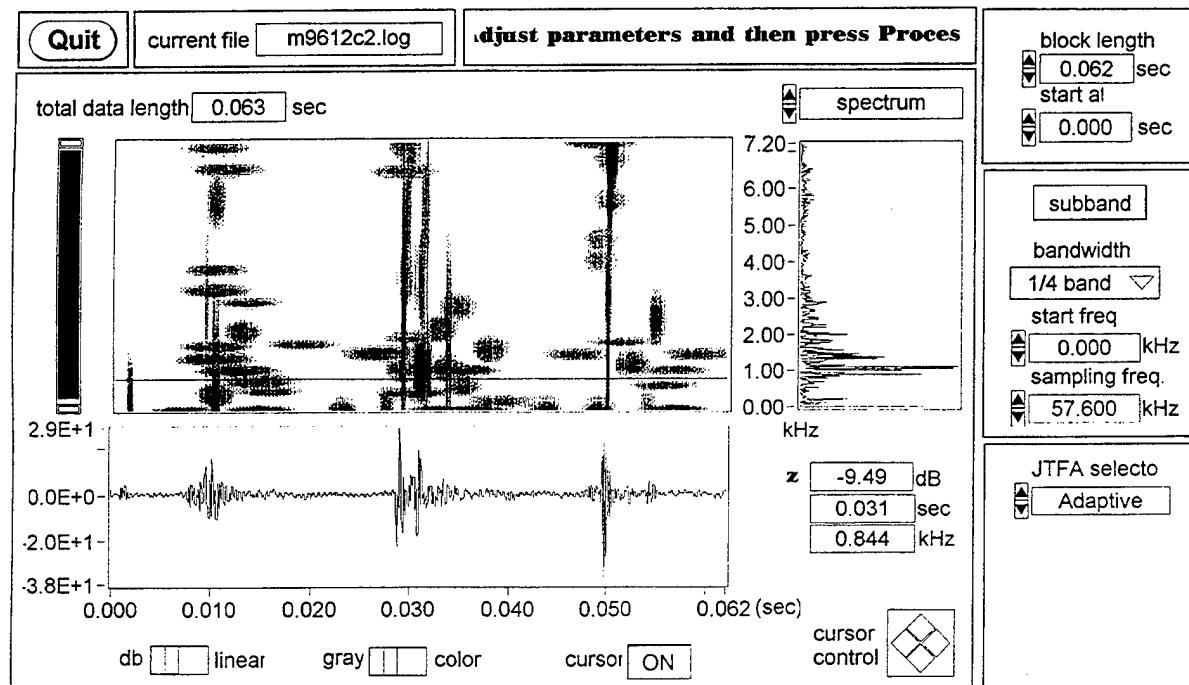


Figure D.31 Head Bolt Transducer C2 950 RPM 125 ft-lb

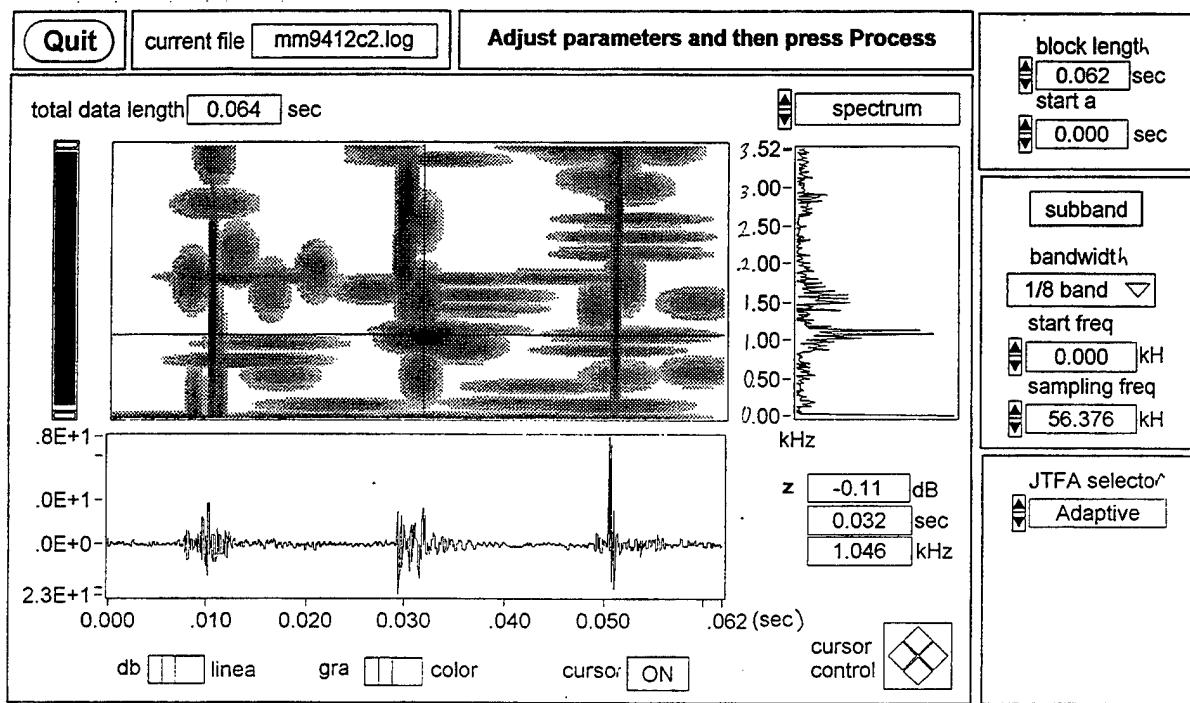


Figure D.32 Head Bolt Transducer C2 950 RPM 125 ft-lb

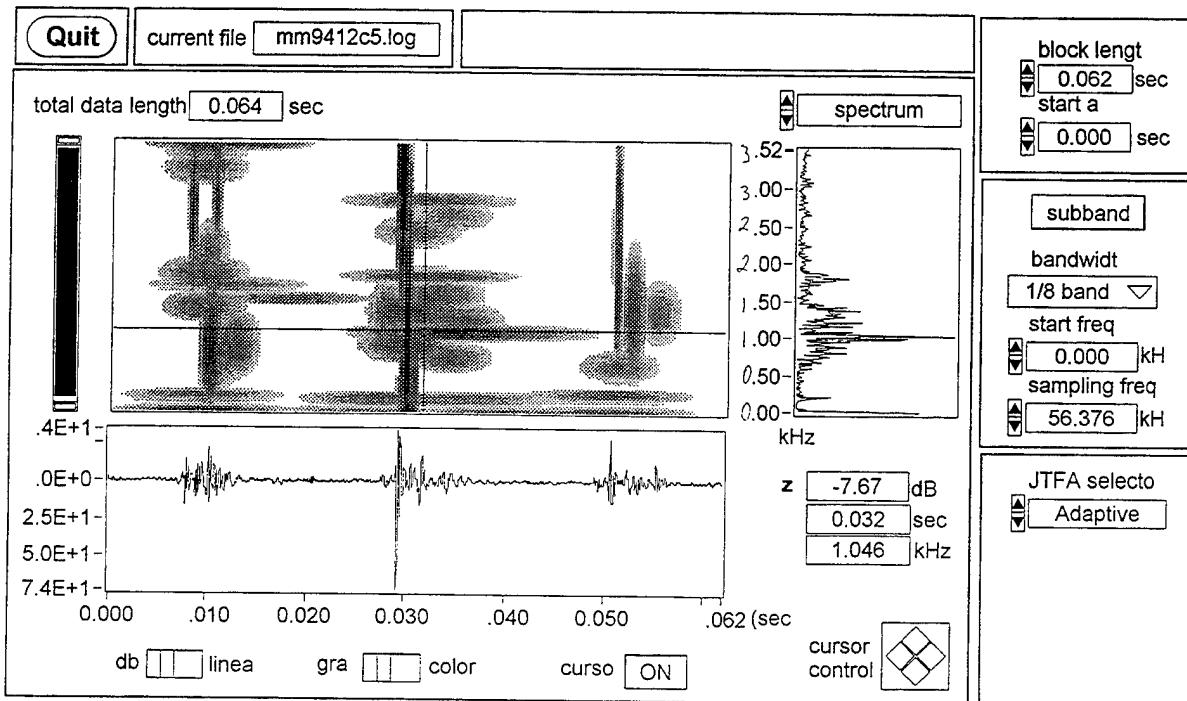


Figure D.33 Head Bolt Transducer C5 950 RPM 125 ft-lb

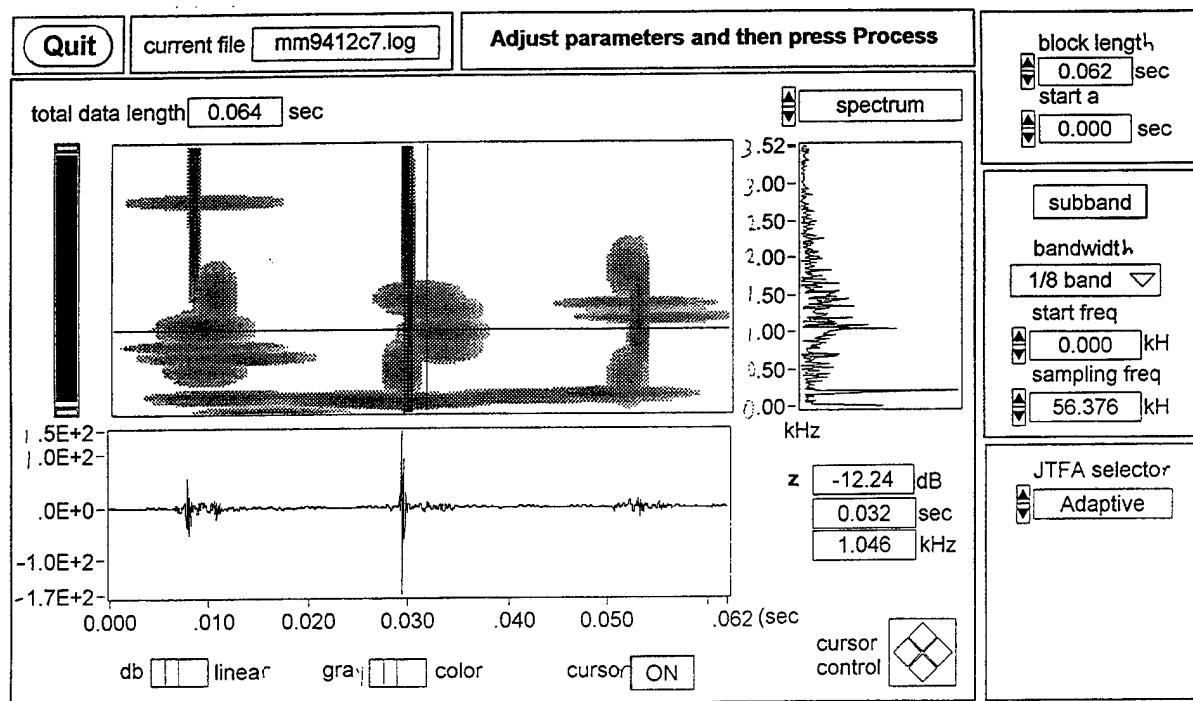


Figure D.34 Head Bolt Transducer C7 950 RPM 125 ft-lb

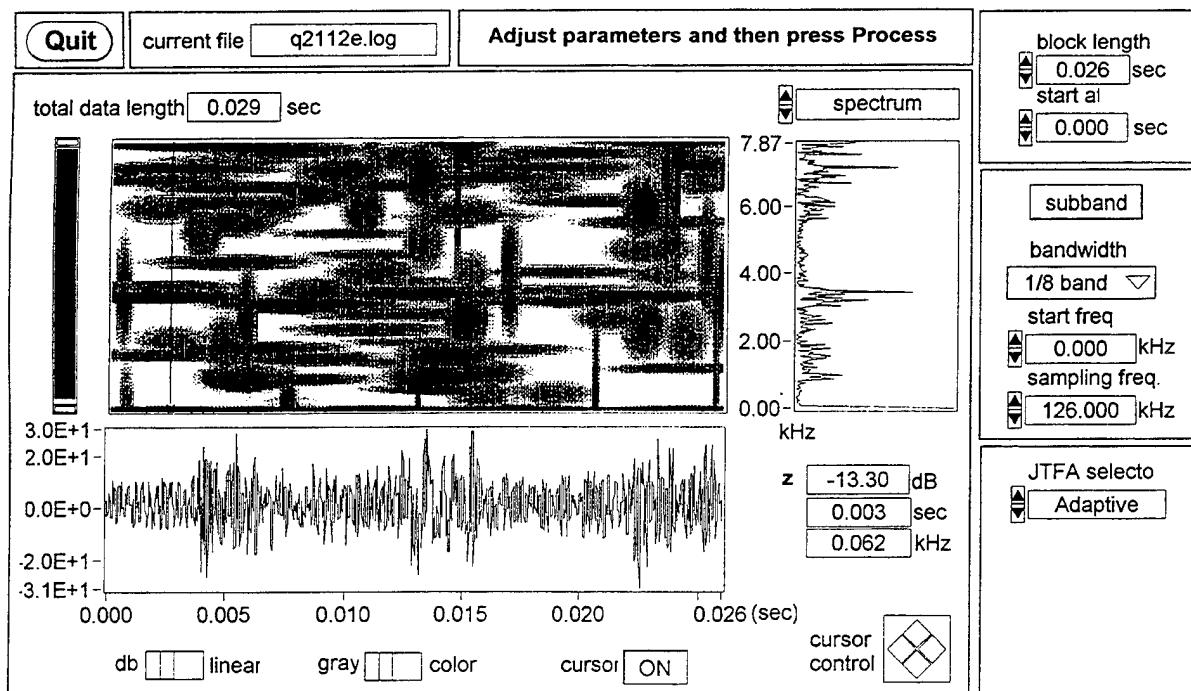


Figure D.35 Engine Transducer 2100 RPM 125 ft-lb

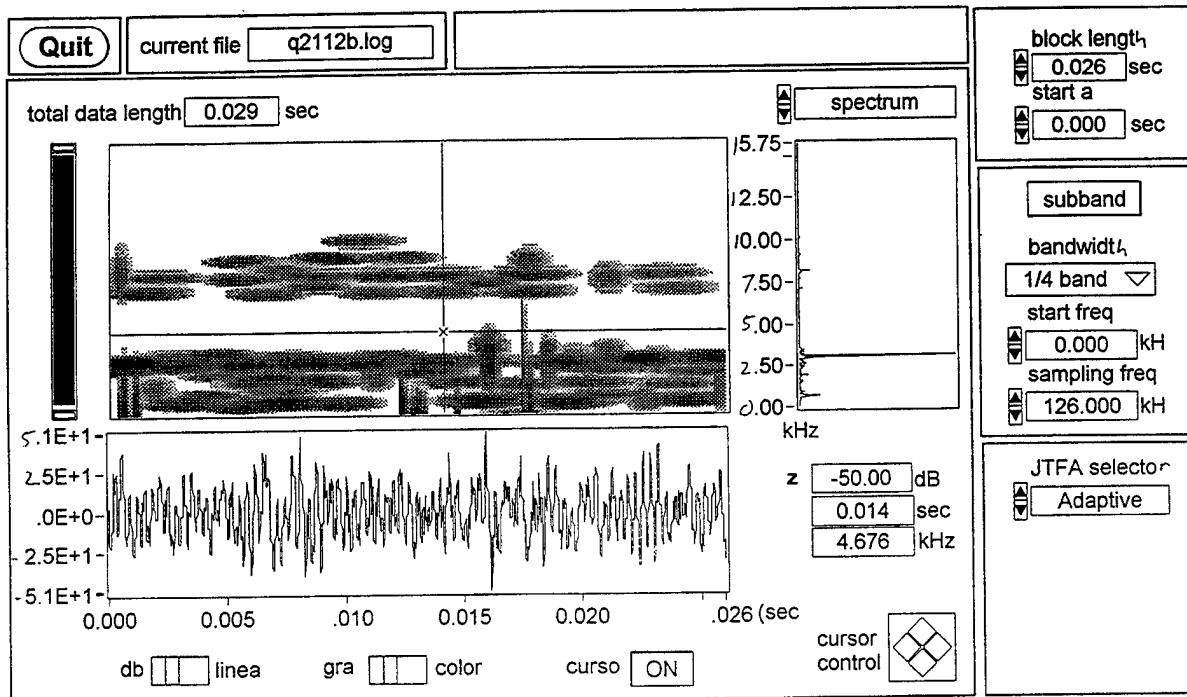


Figure D.36 Engine Base Transducer 2100 RPM 125 ft-lb

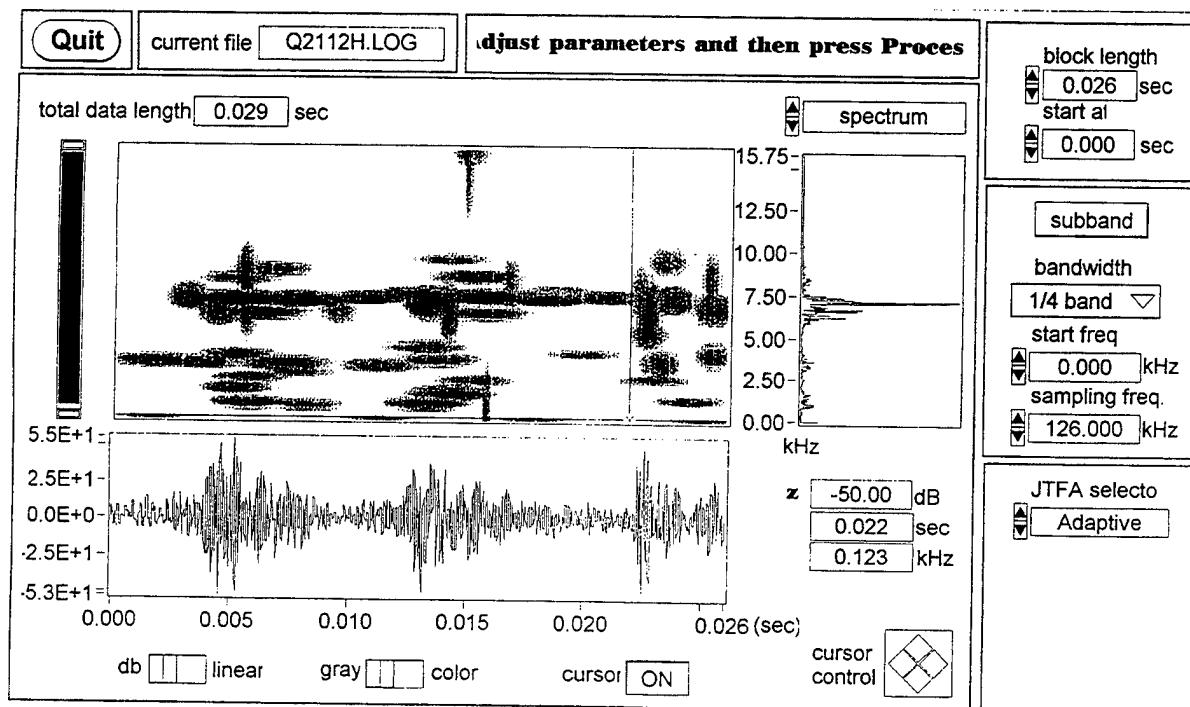


Figure D.37 Head Assembly Transducer 2100 RPM 125 ft-lb

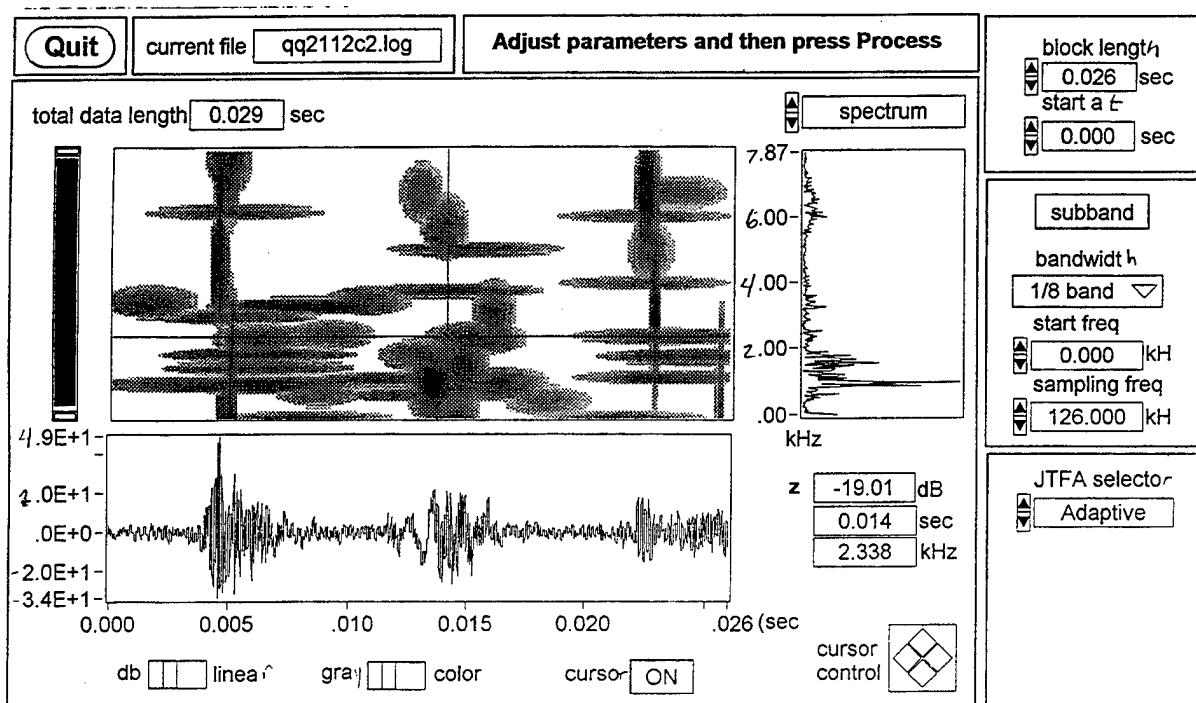


Figure D.38 Head Bolt Transducer C2 2100 RPM 125 ft-lb

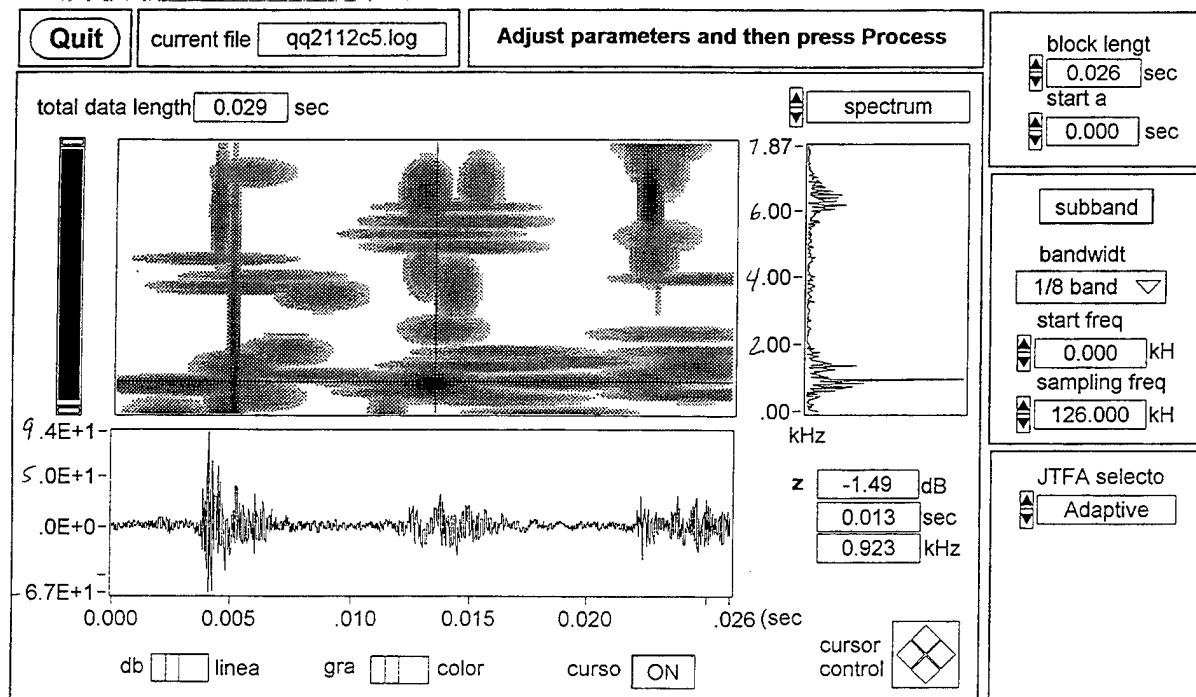


Figure D.39 Head Bolt Transducer C5 2100 RPM 125 ft-lb

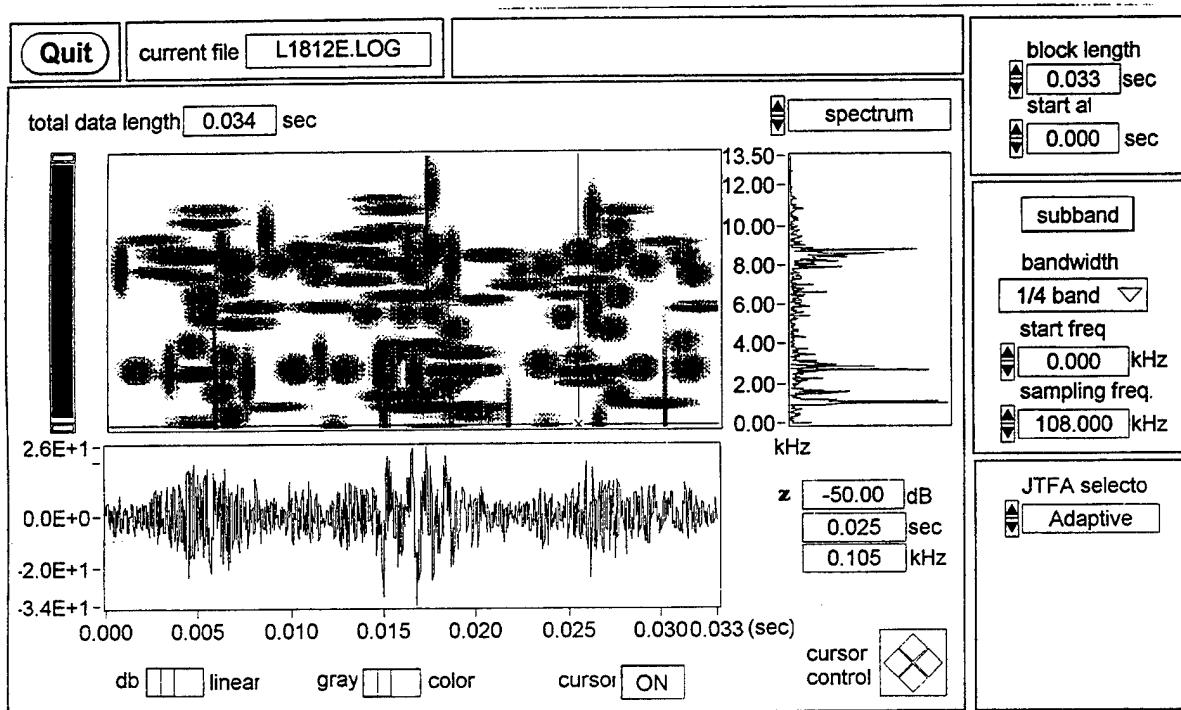


Figure D.40 Engine Transducer 1800 RPM 125 ft-lb

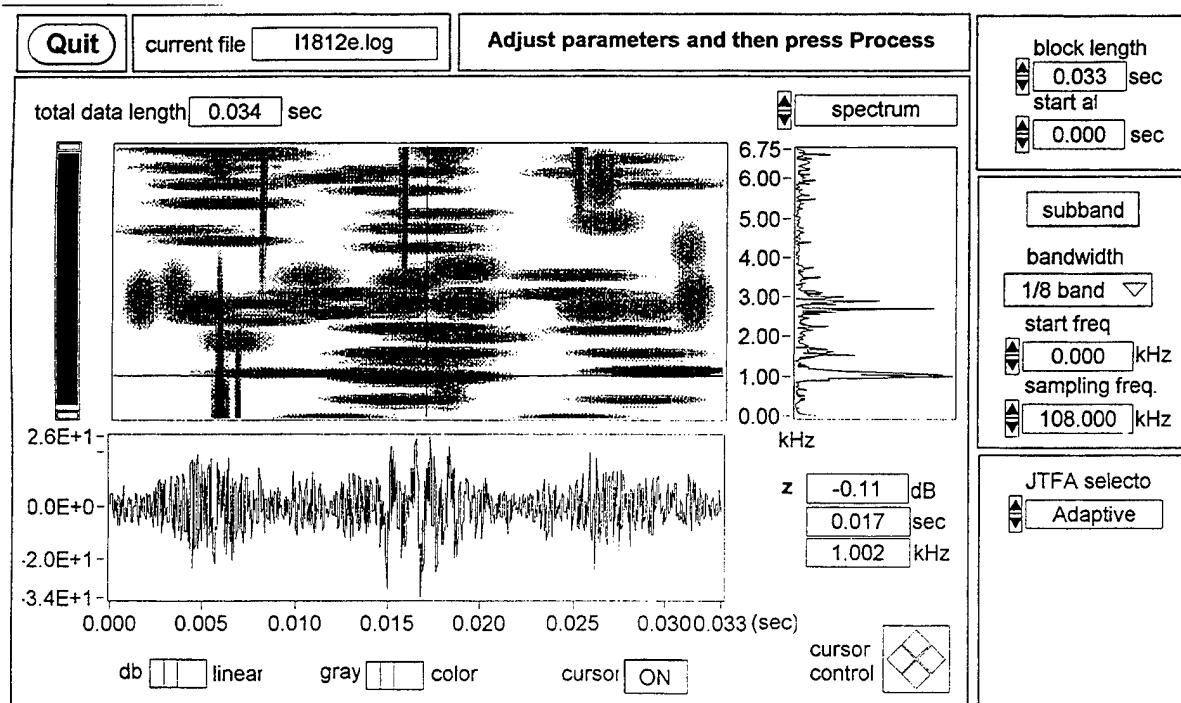


Figure D.41 Engine Transducer 1800 RPM 125 ft-lb

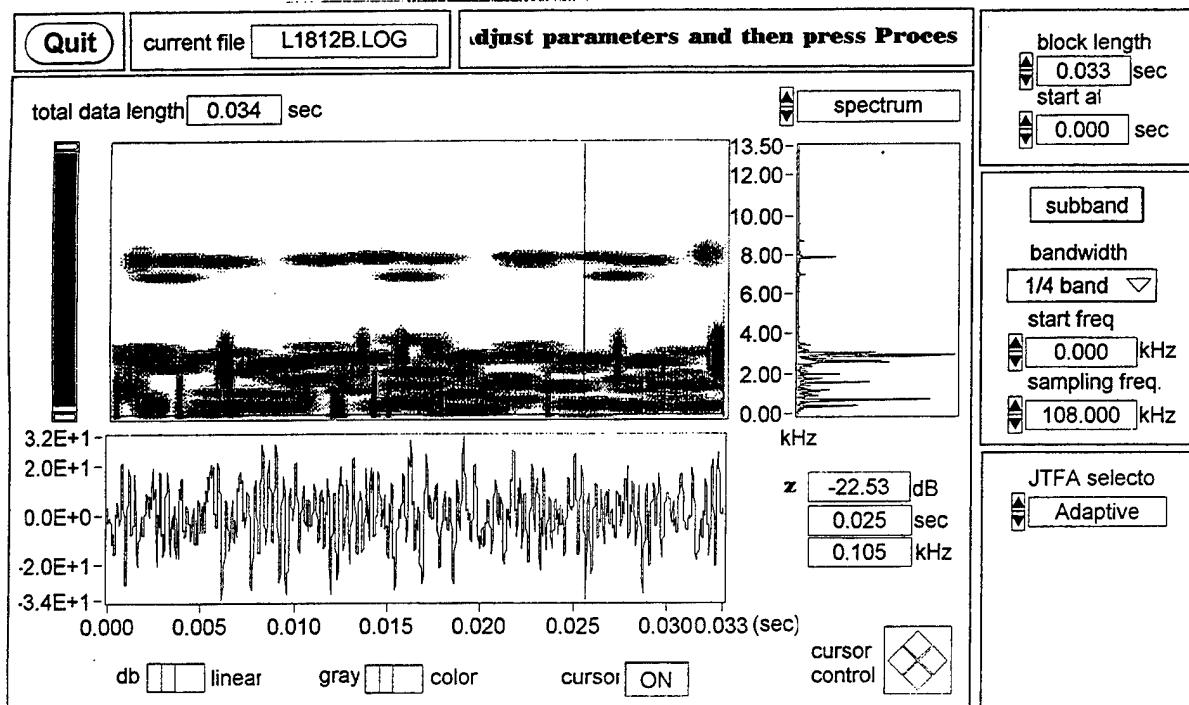


Figure D.42 Engine Base Transducer 1800 RPM 125 ft-lb

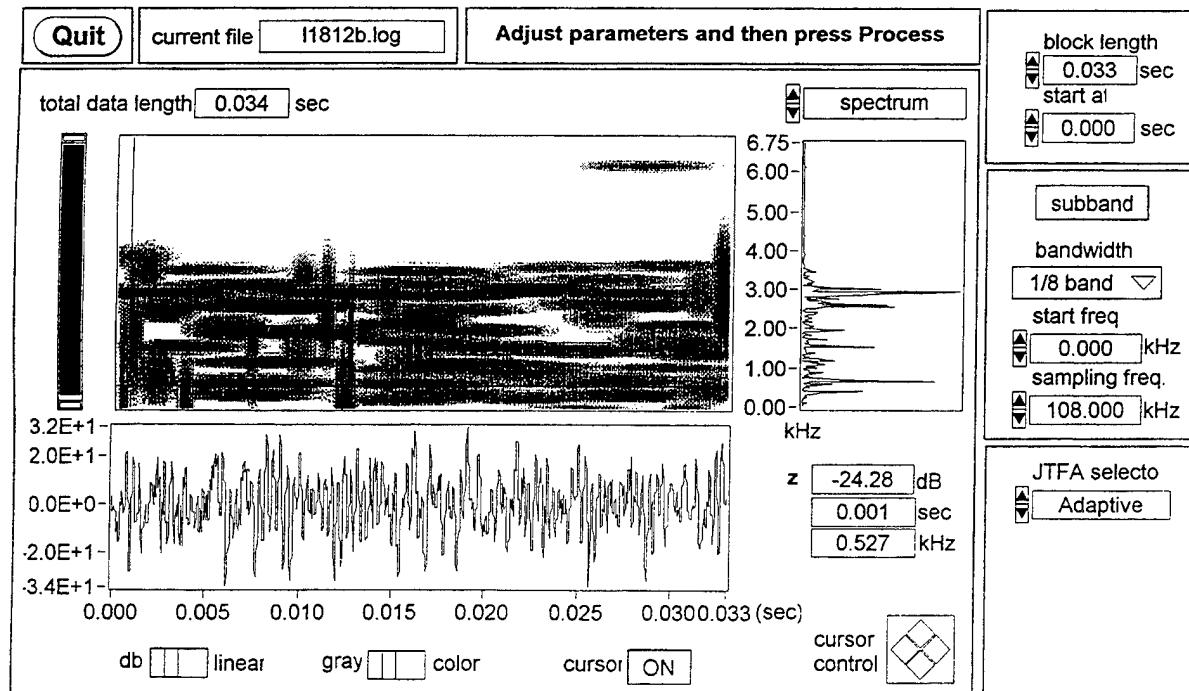


Figure D.43 Engine Base Transducer 1800 RPM 125 ft-lb

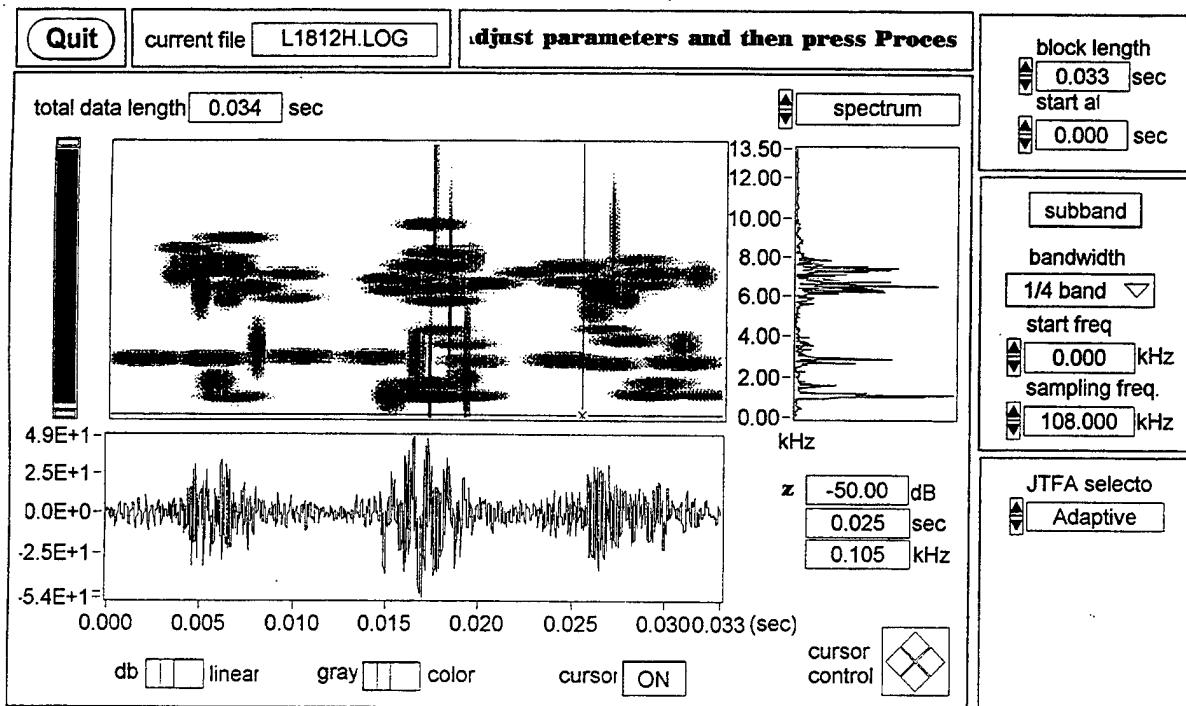


Figure D.44 Head Assembly Transducer 1800 RPM 125 ft-lb

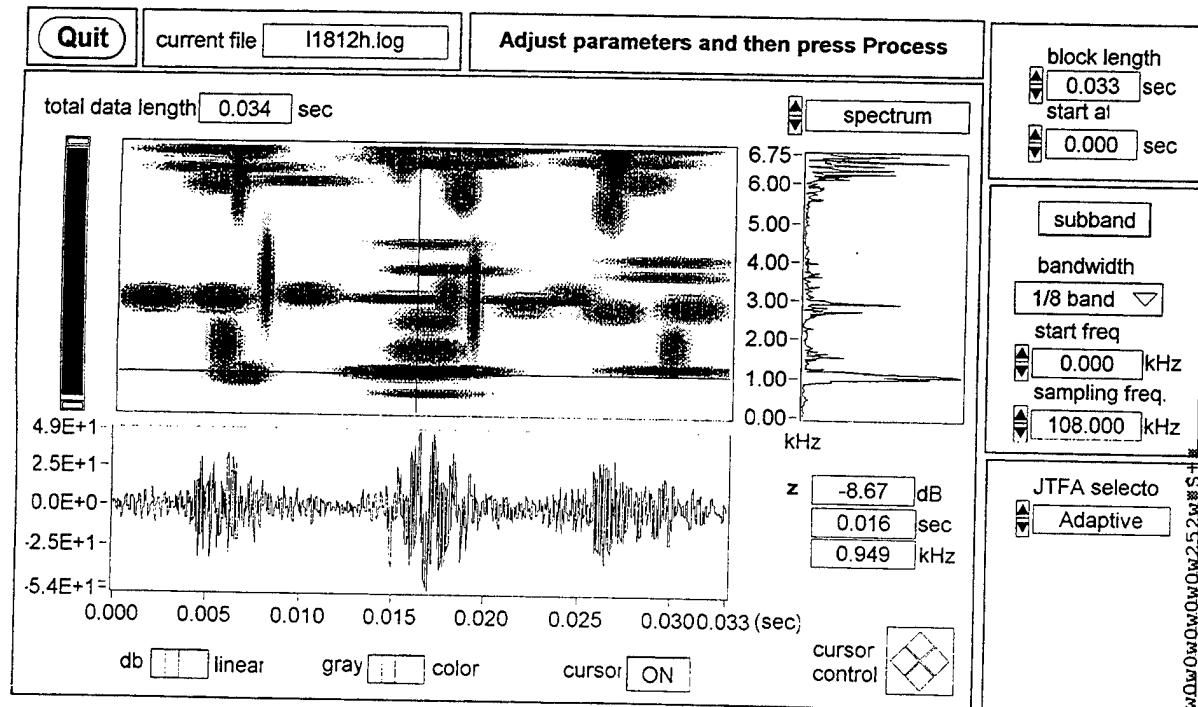


Figure D.45 Head Assembly Transducer 1800 RPM 125 ft-lb

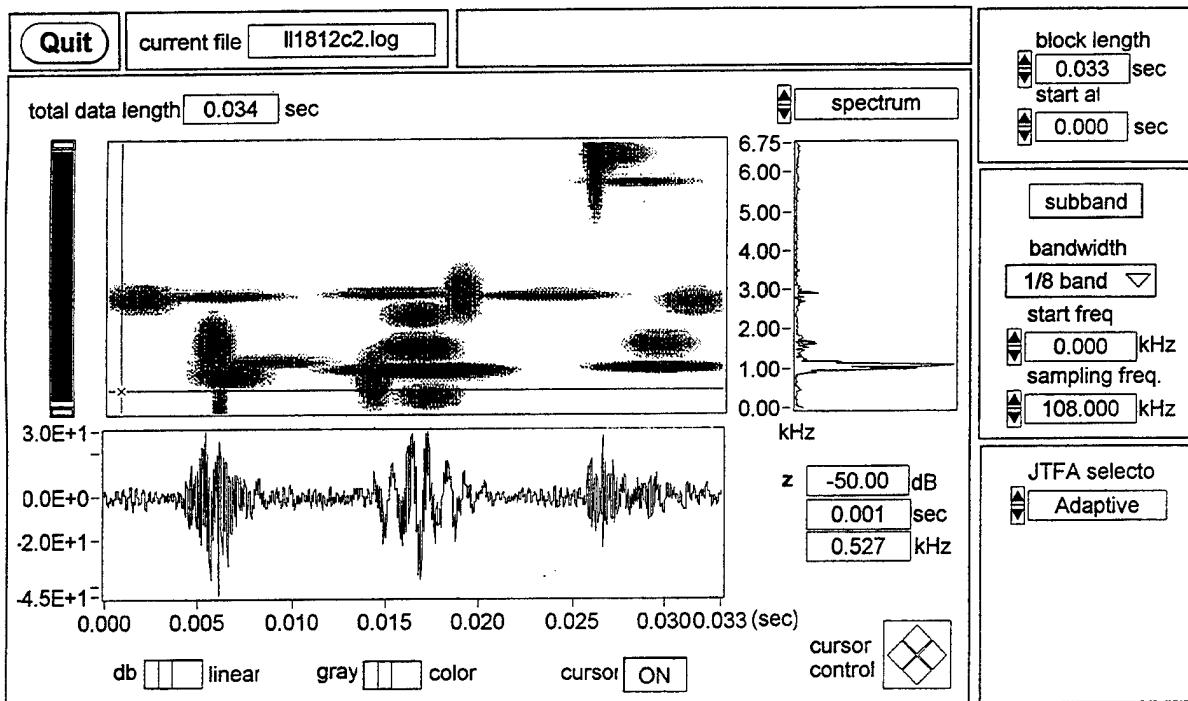


Figure D.46 Head Bolt Transducer C2 1800 RPM 125 ft-lb

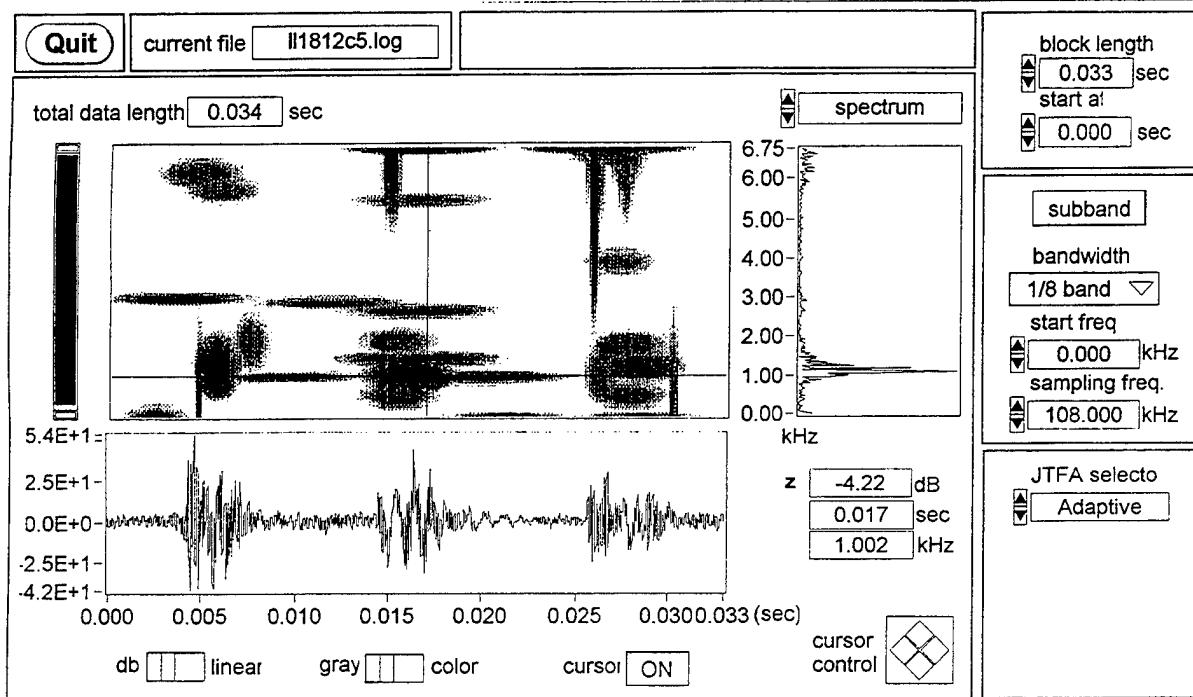


Figure D.47 Head Bolt Transducer C5 1800 RPM 125 ft-lb

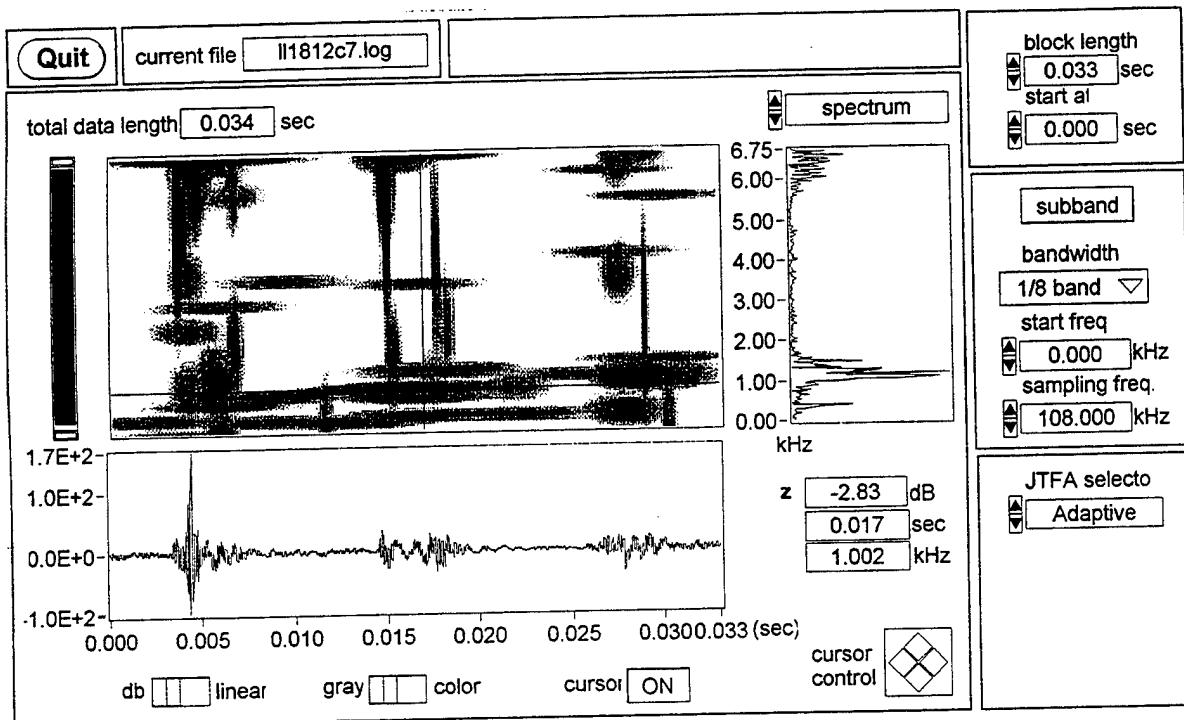


Figure D.48 Head Bolt Transducer C7 1800 RPM 125 ft-lb

APPENDIX E

ECA DATA

The following figures are selected data from the Engine Cycle Analyzer software. The data were taken with the engine at 1800 RPM and under 125 ft-lb torque. The set of data marked BB1812P1.DAT was taken after thirty minutes of the engine running. The data marked CC1812P1.DAT was taken after forty five minutes of run time. These results are typical of the engine under similar circumstances.

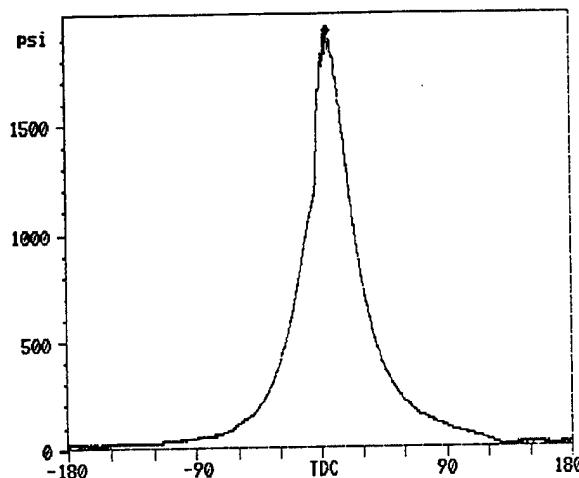


Figure E.1 Pressure vs. Crank Angle
BB1812P1.DAT

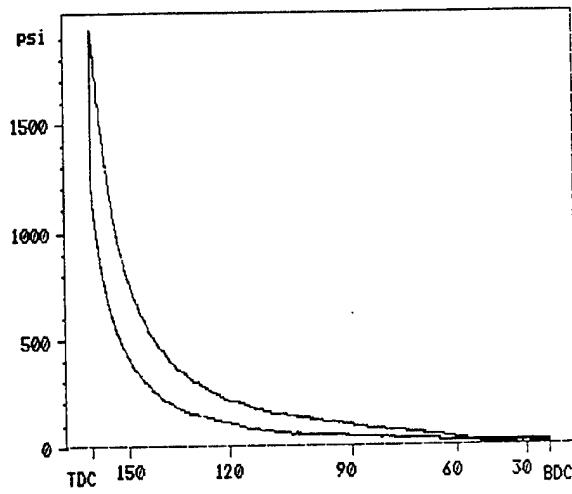


Figure E.2 Pressure vs. Volume
BB1812P1.DAT

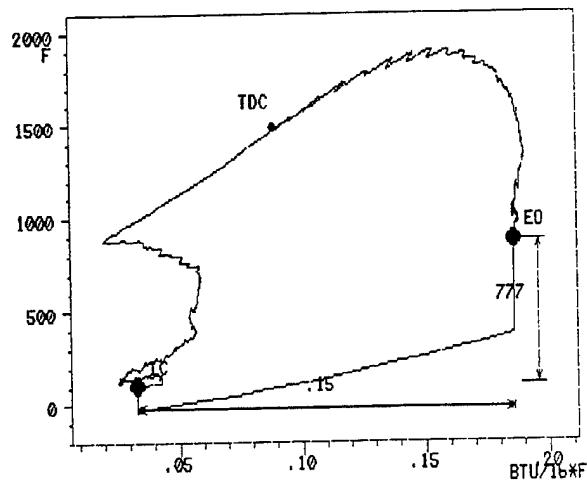


Figure E.3 Temperature vs. Entropy
BB1812P1.DAT

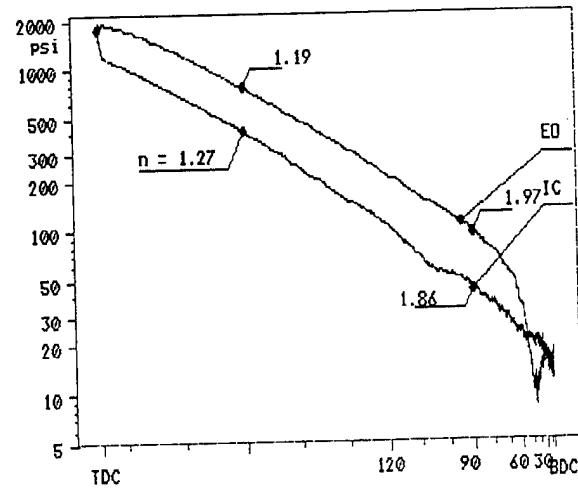


Figure E.4 Log Pressure vs. Log
Volume BB1812P1.DAT

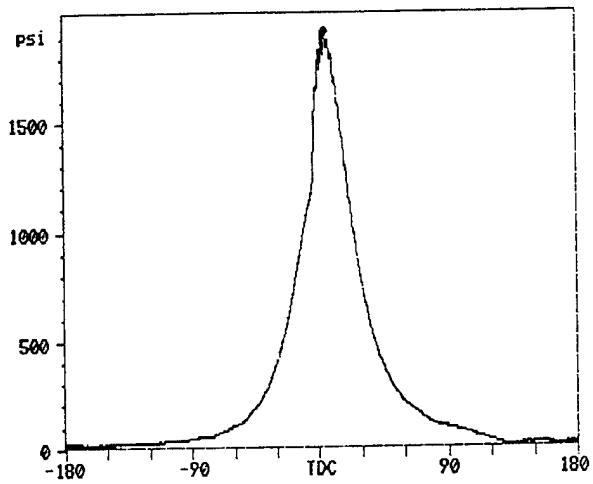


Figure E.5 Pressure vs. Crank Angle
CC1812P1.DAT

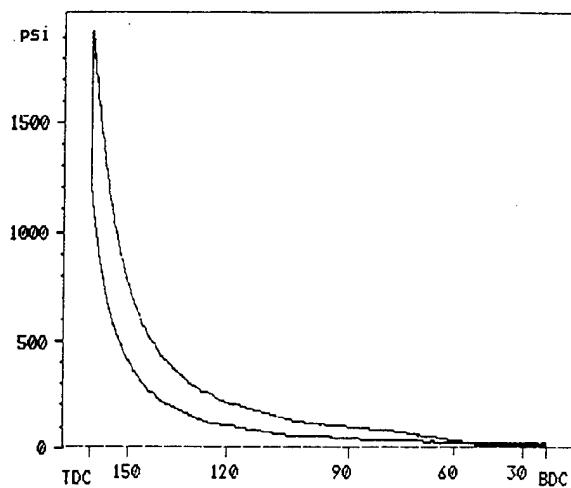


Figure E.6 Pressure vs. Volume
CC1812P1.DAT

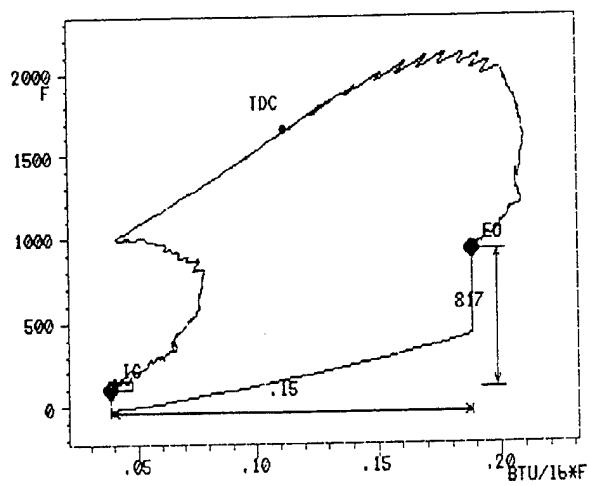


Figure E.7 Temperature vs. Entropy
CC1812P1.DAT

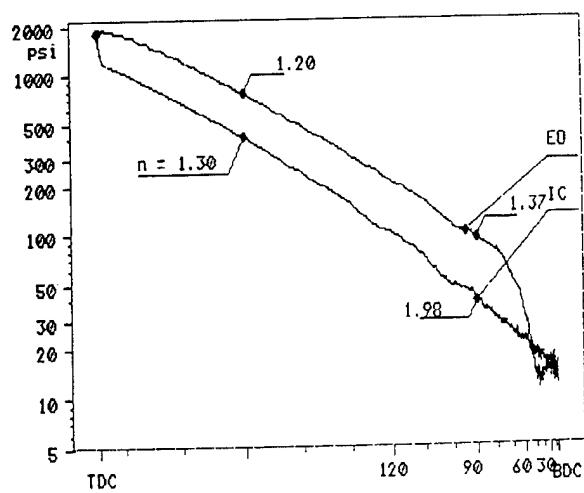


Figure E.8 Log Pressure vs. Log
Volume CC1812P1.DAT

APPENDIX F

VARIATIONS IN SHAFT SPEED DATA

The following are a collection of data collected August 8, 1996 with a Detroit Diesel 53 series three in line cylinder diesel engine. The engine was presumed to be in good working order with no known or imposed malfunctions. The operating conditions are considered to be the same with the exception of varying the torque and engine speed.

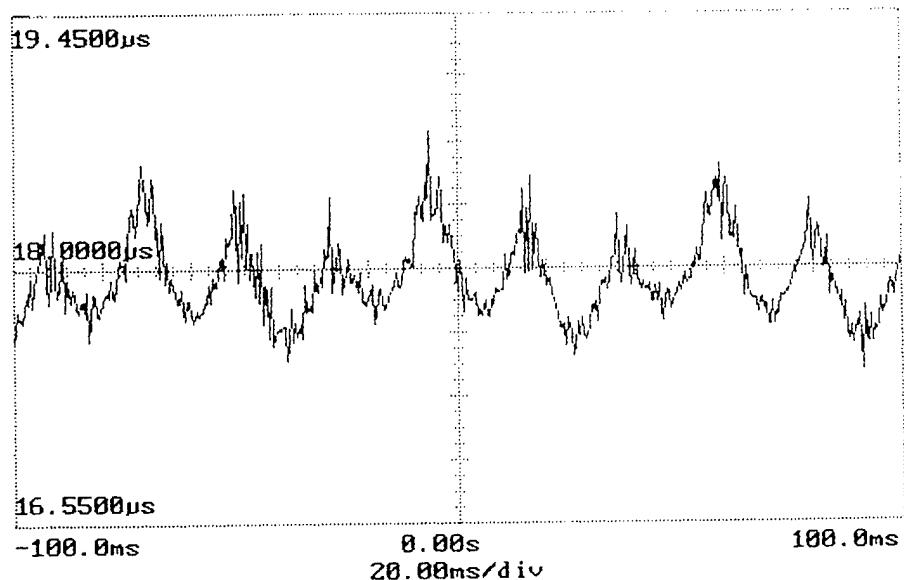


Figure F.1 Shaft Speed Variation 935 RPM 135 lb-ft

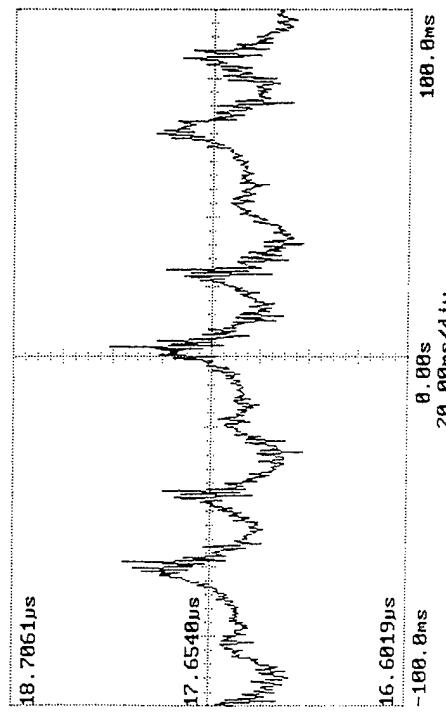


Figure F.2
Shaft Speed Variation
950 RPM 20 ft-lb

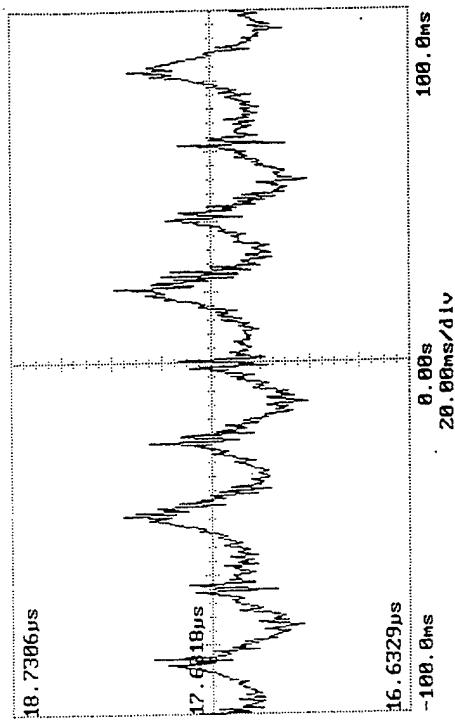


Figure F.3
Shaft Speed Variation
950 RPM 80 ft-lb

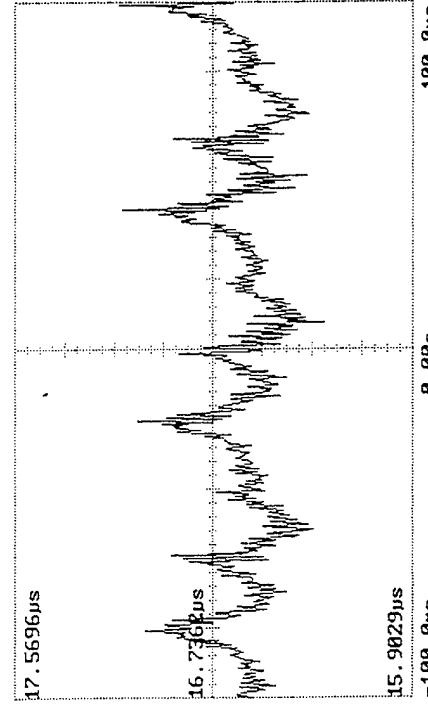


Figure F.4
Shaft Speed Variation
1000 RPM 22 ft-lb

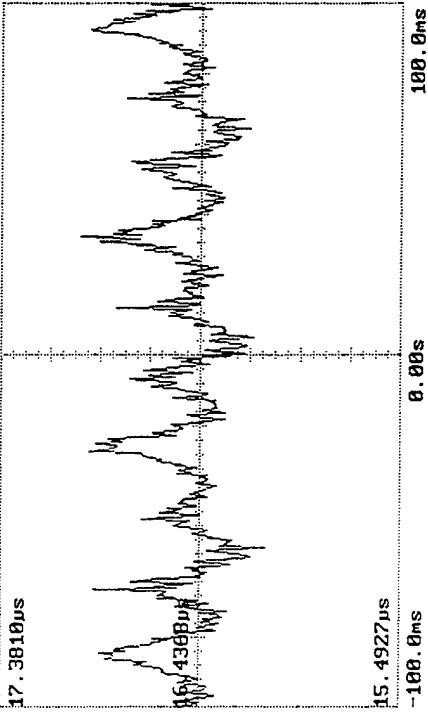


Figure F.5
Shaft Speed Variation
1000 RPM 135 ft-lb

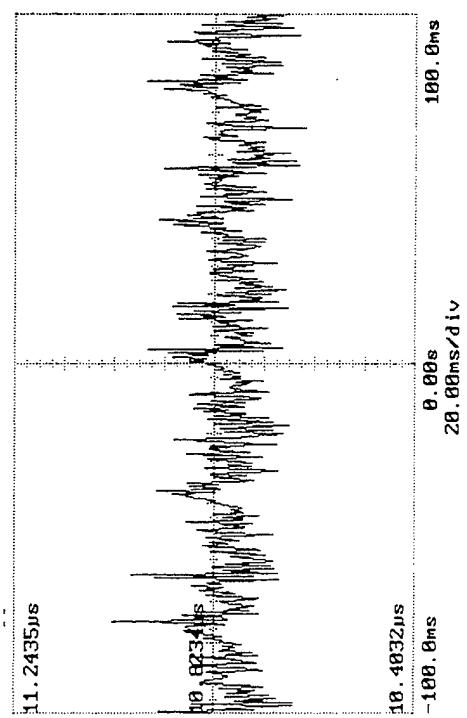


Figure F.6 Shaft Speed Variation
1500 RPM 50 ft-lb

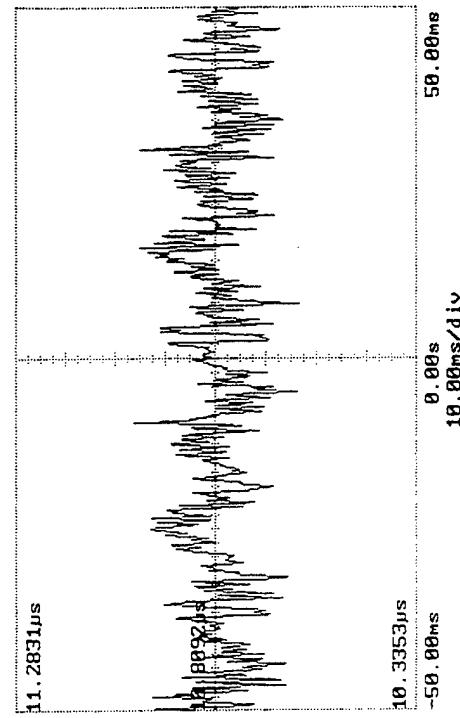


Figure F.7 Shaft Speed Variation
1500 RPM 160 ft-lb

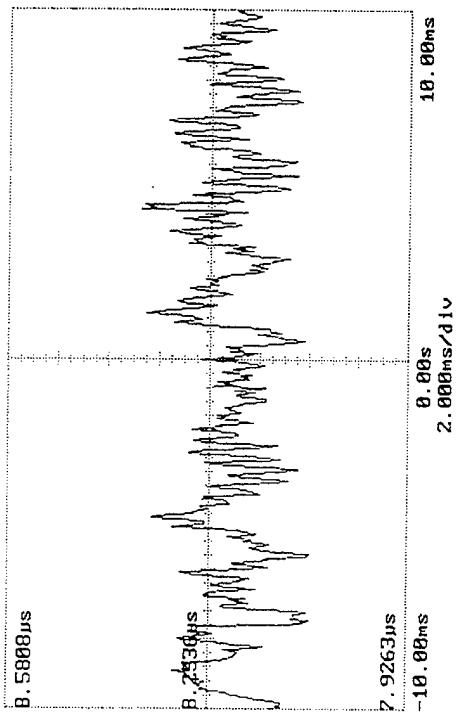


Figure F.8 Shaft Speed Variation
2000 RPM 50 ft-lb

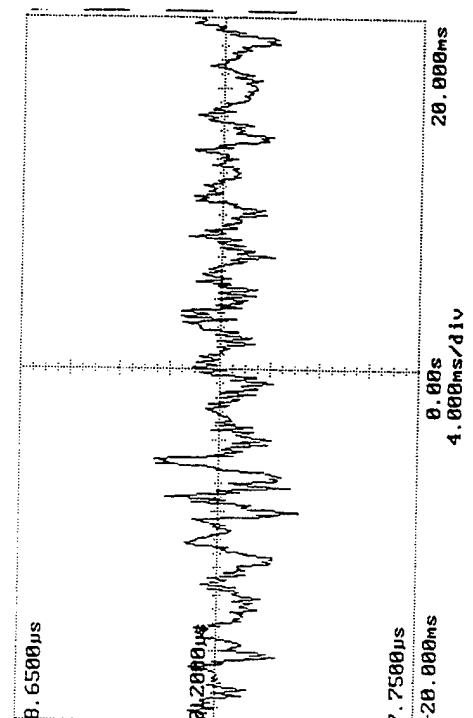


Figure F.9 Shaft Speed Variation
2000 RPM 150 ft-lb

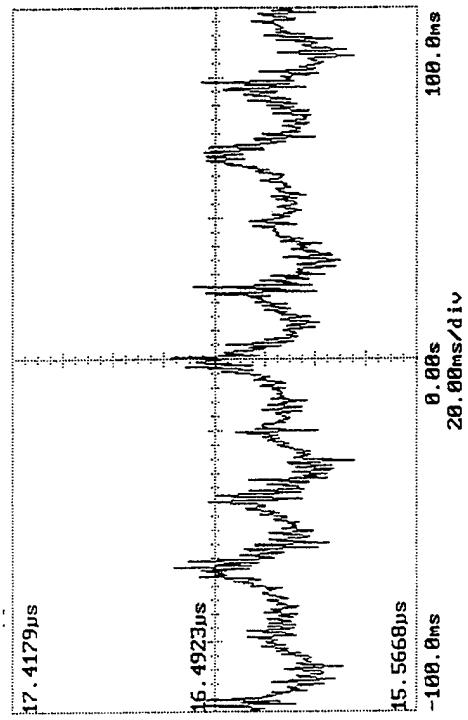


Figure F.11 Shaft Speed Variation
1000 RPM 30 ft-lb
NR 2 Cylinder Glow Plug Loose

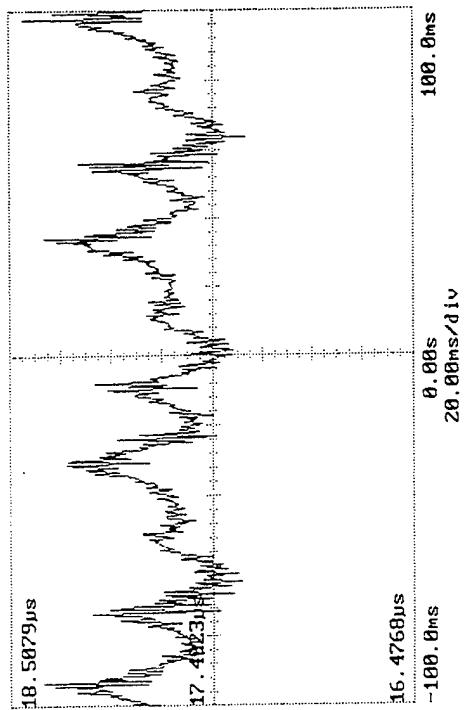


Figure F.10 Shaft Speed Variation
930 RPM 20 ft-lb
NR 2 Cylinder Glow Plug Loose

APPENDIX G

PROCEDURE FOR ESTABLISHING TOP DEAD CENTER

Establishing Top Dead Center (TDC) is a key part of the data acquisition system of the Engine Cycle Analyzer (ECA). It is the reference against which all angular measurements are made. It is the point in the cycle of a cylinder when the volume is the least. The number one cylinder, furthest from the flywheel, was chosen as the cylinder for this reference.

TDC was established by the following procedure after placing the engine out of service:

- a. All of the glow plugs were removed to allow easier rotation of the engine by hand.
- b. A tool was attached to the crankshaft pulley which allowed a large bar to be used to turn the engine over slowly by hand.
- c. The number one cylinder fuel injector and all interferences of the injector tube (the void the injector is seated in) were removed.
- d. A magnetic base was installed on the exhaust manifold. Mounted on this base was a depth micrometer with a 3 inch extension installed. The micrometer was placed in the spray tip hole.
- e. The micrometer shaft should be able to move freely.
- f. The engine was turned over slowly by hand until the micrometer measured a deflection. A mark, point A, was made on the Front Balancer Pulley aligned with a mark on the Engine Front Upper Cover Assembly. (The Front Balancer Pulley was chosen for marking for several reasons. It is readily accessible on the front of the engine; it is not likely to be removed or changed. Lastly, it is geared directly to the crankshaft).

g. The engine was then slowly rotated in the opposite direction until the same deflection was measured with the micrometer. Again a mark, point B, was made on the Front Balancer Pulley as it aligned with the Engine Front Upper Cover Assembly mark.

h. Steps f and g were repeated several times until point A and B did not vary from one rotation to the next.

i. A chord connecting points A and B was drawn on the Front Balancer Pulley. The midpoint of this chord will be the radial of TDC. Using a drafters square a line perpendicular to the chord was drawn. This point is the location of TDC.

j. The micrometer was pulled out of the engine approximately one inch to allow the full stroke of the piston to be completed without exceeding the range of the micrometer.

l. The engine was rotated by hand and the maximum deflection of the micrometer was noted through several rotations. Most of the time this TDC coincided with the mark from step i.

m. A permanent mark was place on the engine and the balance wheel to denote TDC.

n. The engine was returned to operational.

The technique described above was used instead of just doing step L for the accuracy of the measurement. Measuring the piston location before the piston is at TDC means that for a small rotation the displacement of the piston will be greatest. When the piston is at TDC, a large change in angular rotation results in very little noticeable displacement of the piston and would result in an inaccurate reading.

APPENDIX H

RELATIVE MOTION BETWEEN ENGINE COMPONENTS

The vibration signals from the various major engine assemblies were compared against one another to determine if there was any relative motion. The plots were created in MATLAB and are simply subtracting the signal of one engine part from another engine part.

The expectation of this comparison was that there would be little difference between the head bolt and the head assembly, but that was not the case. Similarly, we expected there to be little motion between the base and the engine, again we were surprised. We also expected to see some relative motion between the head assembly and the engine. This did occur and it followed the same relative pattern of combustion seen when examining individual head bolt vibrations.

Relative motion between the head assembly and the engine was expected because we believe the head to be move like a body mounted to another body, the engine block, by springs, the cylinder head bolts.

No further analysis was performed on these results.

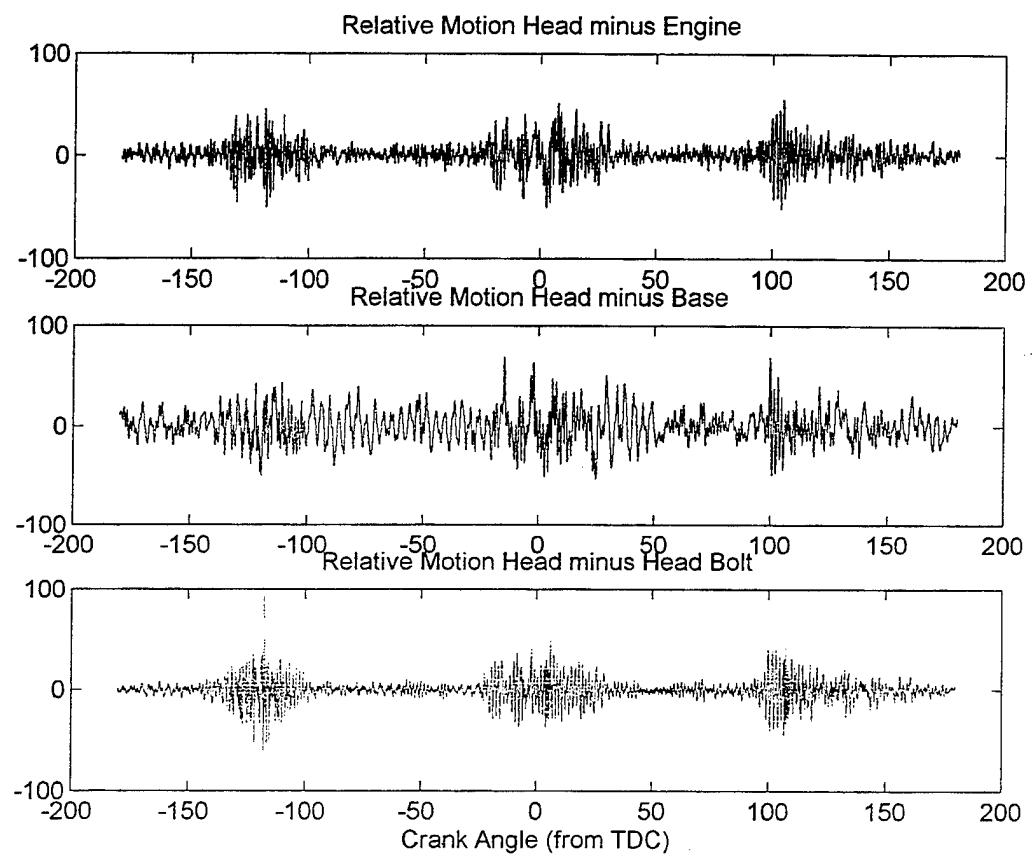


Figure H.1 Relative Motion between Cylinder Head Assembly and Engine Parts vs. Crank Angle

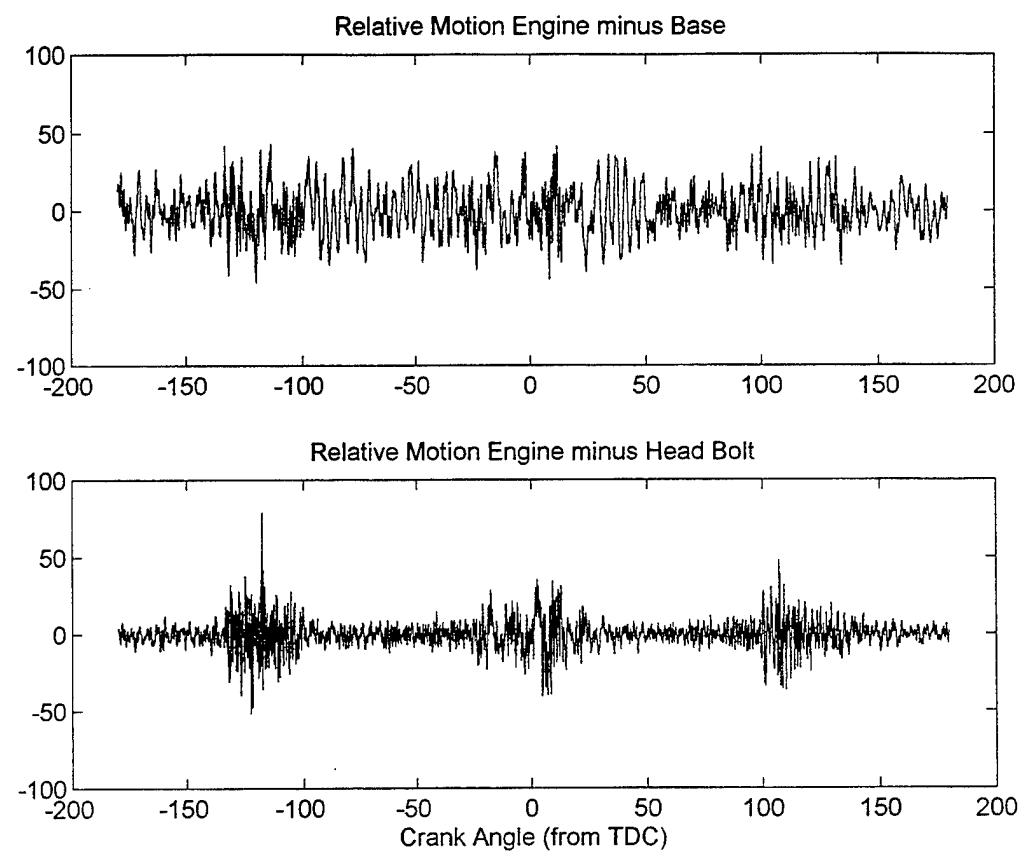


Figure H.2 Relative Motion between Engine Block Assembly and Other Engine Parts vs. Crank Angle

APPENDIX I

DATA FILE NOMENCLATURE AND MANAGEMENT

The ECA collects data each time the optical encoder sends a signal. This results in 3600 points of data for each cycle. The ECA software saves these points for a requested number of cycles, optimally 100, and allows them to be saved individually or averaged to a single file. All data collected was saved to the hard drive under the DATA directory. This directory contains many subdirectories, each one the date which the data was collected. The file names indicate the type of run and the sensor used. A sample file name is A2010P1.DAT. This file name breaks down in the following way:

- A- Any letter used to separate data runs
- 20- These numbers represent the RPM. This means the data was taken at an engine speed of 2000 RPM.
- 10- This number indicates the torque on the engine. This was at 100 ft-lbs.
- P1-indicates the sensor used. P for pressure sensor. C for a Head bolt vibration transducer. The number following indicates the particular sensor. The remaining are vibration transducers on the following parts: E for engine block. H for cylinder head assembly. B for engine base.
- .DAT- This file extension is an ECA convention. DAT is an Engine Cycle File. .TDC is a Hot Motoring function. These are both files of averaged data. A file extension of a number (ie. A2014C2.27) is an individual cycle of data and has not been averaged with any other files.

Prior to collecting data the user should create a new subdirectory in the DATA directory which will distinguish the newest data from older data. The directory convention used was the date the data was collected (ie. C:\data\aug0596).

The data files are saved in ASCII format with a header and a footer which can be read by the Joint Time Frequency Analyzer Software but not by MATLAB. For use with MATLAB, the header and footer must be removed. The resulting data file is a single column 3600 numbers long.

Data file conversion may be accomplished in a word processor. The resulting file must be saved in ASCII for MATLAB or the JTFA to read them.

Other functions of the ECA are described in detail in Section IV A and reference 3.

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